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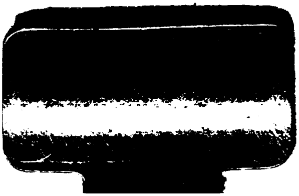

~~Kurt F. Wendt~~

~~University of Wisconsin~~

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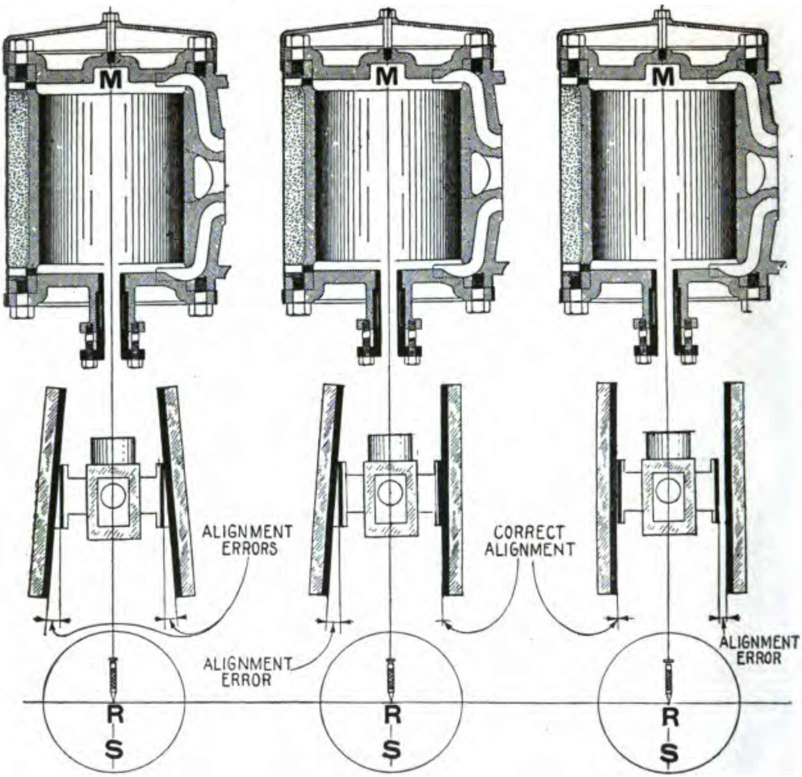

Plate 3.—Errors in Engine Alignment

Cross head guides not parallel.

Since only one gib of the cross head bears on the guide at a time, the tendency of the error is to cause the piston to bind at the cylinder ends, or if much out, to cause the cross head to bind at one end.

One guide not parallel to cylinder axis.

Where adjustable flat guides are used as on some marine engines, an error in parallelism of one guide may be sometimes detected by faulty running in forward or reverse motion according to which guide is out of alignment.

Guides at different distances from cylinder axis.

If the far guide be opposite the working gib, it will cause *slapping*, that is, a knock during admission.

In the figures **MS,** is the cylinder axis; MR, plumb line through center of cylinder end.

AUDELS ENGINEERS AND MECHANICS GUIDE 3

A PROGRESSIVE ILLUSTRATED SERIES
WITH QUESTIONS-ANSWERS
CALCULATIONS

COVERING

MODERN ENGINEERING PRACTICE

SPECIALLY PREPARED FOR ALL ENGINEERS
ALL MECHANICS AND ALL ELECTRICIANS.
A PRACTICAL COURSE OF STUDY AND
REFERENCE FOR ALL STUDENTS AND
WORKERS IN EVERY BRANCH OF THE
ENGINEERING PROFESSION

BY
FRANK D. GRAHAM, B.S., M.S., M.E.
GRADUATE PRINCETON UNIVERSITY
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STATIONARY AND MARINE ENGINEER

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NOTE

In planning this helpful series of Educators, it has been the aim of the author and publishers to present step by step a *logical plan of study in* ***General Engineering Practice,*** taking the middle ground in making the information readily available and showing by text, illustration, question and answer, and calculation, the theories, fundamentals and modern applications, including construction *in an* **interesting** *and* **easily understandable** *form.*

Where the question and answer form is used, the plan has been to give *short*, *simple* and *direct answers*, limited to one paragraph, thus simplifying the more complex matter.

In order to have adequate space for the presentation of the important matter and not to divert the attention of the reader, descriptions of machines have been excluded from the main text, being printed in smaller type under the illustrations.

Leonardo Da Vinci once said:

> "Those who give themselves to ready and rapid practice before they have learned the theory, resemble sailors who go to sea in a vessel without a rudder"

—in other words, "*a little knowledge is a dangerous thing.*" Accordingly the author has endeavored to give ***as much information as possible*** in the space allotted to each subject.

> The author is indebted to the various manufacturers for their co-operation in furnishing cuts and information relating to their products.

These books will speak for themselves and will find their place in the great field of Engineering.

Contents of Guide No. 3

CHAPTER PAGES

37. Locomotives........................ 951 to 1,044

Historical—the modern locomotive—*classification*—the boiler—types—construction—grate area and heating surface—smoke box—dry pipe and throttle valve—boiler attachments—cylinders—valve gear—piston and rod—guides—cross head—connecting. rods—valve gear—**Compound Locomotives**—running gear—tender—***air brake***—names of parts—operation—the various "positions" explained—double heading—feed valve—compressors—dead engine device—*operating instructions*—COMPRESSOR TROUBLES.

38. Marine Engines.................. 1,045 to 1,090

Types—characteristics of marine loads—walking beam engines—***valve setting***—inclined paddle engines—stern wheel engines—variable cut off valve—cam drive—stern wheel boat proportions—vertical engines—simple engines—jackets—author's system—***compound engines***—engine of steamer ***Stornoway I***—steeple type—***triple expansion engines***—author's single acting triple engine—graphical method of proportioning cylinders—combined diagrams—cards—various pump drives—piping—***quadruple expansion engines***—operating conditions—various cylinder adjustments—***quintuple expansion engine*** author's diagrams for 1,000 and 1,500 lbs. pressure.

39. Installation....................... 1,091 to 1,114

Location—*proper arrangement of exhaust*—foundations—placing the main castings—alignment—grouting—assembling the moving parts—marine engine shaft alignment.

40. Lubricants........................1,115 to 1,122

Importance of proper lubricants—desirable qualities of a lubricant—cold, flash and burning points—classes of lubricants—***choice of a lubricant.***

41. Lubrication........................1,123 to 1,142

How to oil an engine—internal lubrication—gravity and hydrokinetic lubricators—***practical points***—force feed—external lubricating systems.

42. Practical Management..........1,143 to 1,192

Duties of the engineer—1. OPERATION—before starting—starting with: *a*, direct connected air pump, *b*, jet condenser, *c*, siphon condenser; *d*, exhaust steam condenser—running—lubrication—points on belts—knocks—hot bearings—feed pump—detail of steamer ***Stornoway I*** machinery—stopping with: *a*, direct air pump, *b*, jet condenser; *c*, siphon condenser; *d*, exhaust steam condenser—2. CARE—packing—methods of applying packing—use and abuse of the Moncky wrench—various wrenches—calipering—4. LAYING UP—rust preventives—drainage—sweating—air conditions.

43. Rotary Engines...................1,193 to 1,200

Misdirected efforts of inventors—early rotary engines—***non-expansion and expansion types***—reversing methods.

44. Steam Turbines..................1,201 to 1,268

Adaptation—***principles***—nozzles and flow of steam—parallel and diverging nozzles—***classification of turbines***—impulse and reaction—simple and compound ***impulse turbines***—simple multi-stage turbines; pressure velocity diagram—construction details—compound multi-stage impulse turbine—construction details—***reaction turbines***—sectional view of Parson's turbines showing construction—Terry turbine—

Steam Turbines—*Continued*

—***turbine governors:***—throttling, partial admission, intermittent admission, by pass types—details of construction—Curtiss turbine governors—***working pressures for turbine:***—high, low, mixed—Terry compound two stage radial flow turbine—turbine driven dynamo for locomotive head light—blading of Allis Chalmers Parsons type turbine—***low pressure turbines:***—exhaust steam operation—vacuum limit for reciprocating engines—Rateau accumulator—accumulator accessories—gland water purifier—oil piping—importance of high vacuum—Curtiss compound multi-stage turbine—step bearing—Curtis forced lubrication system—Curtis emergency governor, mechanical valve gear, upper stuffing, self-centering ring, by pass or stage valve—***starting a Curtis turbine***—method of determining clearance—construction details—combined reciprocating marine engines and turbines—***running a Parsons turbine***—Allis Chalmers Parsons turbine construction details—Buffalo-"spiro" turbine—***De Laval operating instructions***—reduction gear—De Laval construction details.

45. **Indicators**.........................1,269 to 1,314

Description and operation—explanation of diagram—parallel motion—springs—preparing the indicator for use—reducing motions—indicator piping—***how to take a card***—the diagram—***finding the m.e.p.***—electric pencil control—***theoretical water consumption calculation***—ordinate scale—planimeter—indicating gas engines—characteristic diagrams—indicating compound and triple engines

CHAPTER 37

LOCOMOTIVES

The first locomotive built in this country was designed and constructed by Peter Cooper, in 1829, and in the following year was successfully run on the Baltimore and Ohio Railroad.

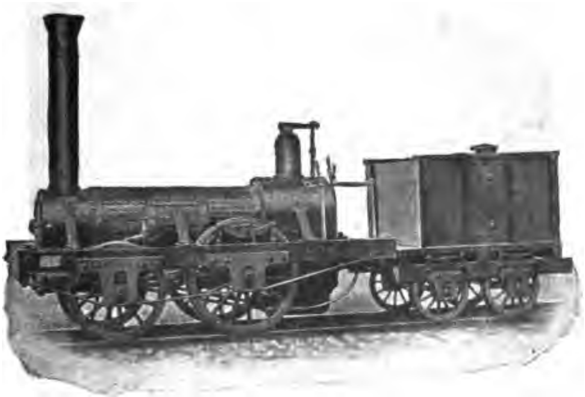


FIG. 1,845.—***"Old Ironsides."*** This engine was built by Baldwin for the Camden and Amboy railroad, and tried on the road November 23, 1832. The principal dimensions are: driving wheels, 54 ins.; front wheels, 45 ins.; cylinders, 9½×18; boiler, 30 ins. in diam., 72—1½″×7′ copper tubes. The valve motion had at first loose eccentrics, which were later replaced by fixed eccentrics. According to the account of the trial trip as given in The Philadelphia Chronicle, "the placing fire in the furnace and raising steam occupied 20 minutes, and the engine (with her tender) moved from the depot in beautiful style, working with great ease and uniformity."

This engine was only a working model, and not intended for permanent service. It had only one cylinder, 3¼ inches in diameter, and weighed less than one ton. A speed of eighteen miles an hour was obtained with Peter Cooper as engineer and a load of forty passengers.

The DeWitt Clinton, built at West Point in 1831, for the Mohawk and Hudson Railroad, weighed three and one-half tons, and was fitted with 5¼ × 16 inch cylinders and 54 inch drivers. The boiler was of the horizontal type.

The first locomotive built by Matthias and Baldwin, of Philadelphia, was "Old Ironsides," as shown in fig. 1,845; it was constructed in 1832 for the Philadelphia and Germantown Railroad. This engine was fitted with the hook motion valve gear, which was generally used many years after the invention of the shifting link. The latter became the standard for locomotives about 1856, and its adoption marks a dividing line between the present and early transitional stages of locomotive practice.

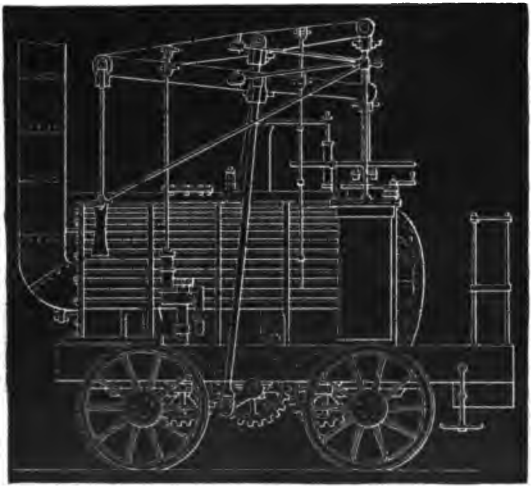


FIG. 1,846.—Hedley's "Puffing Billy," first locomotive used commercially in hauling cars. This engine, which was built in 1813, had a return flue boiler, which provided all the steam required. This form of boiler which had been used by Trevithick and Hedley made it a practical success. It was the best form of boiler used until Robert Stephenson applied the multi-tubular boiler to the "Rocket" in 1829. The "Puffing Billy" had a furnace extending about half way into the boiler and a flue leading to an up-take, from which the gases of combustion passed through a return flue to the smokestack. The fireman did his work at the smokestack end of the boiler and the engineer sat in front on a wooden seat held by four upright iron posts. The frames are of wood, quite substantial in form and rest upon the axles without the intervention of springs. The four wheels are connected by inside gearing, and the motion is transmitted to a gear wheel upon a separate axle. The tank is an oblong iron box, set behind the coal space.

The Modern Locomotive.—In present day construction, the locomotive consists of a horizontal boiler and two engines attached to a frame, which, in turn, is supported by suitable running gear, a tender being coupled behind to carry a supply of fuel and water.

The weight of the engine does not rest directly upon the axle bearings, but is carried by several springs, thus forming a cushion to protect the heavy parts from too much vibration and jar, due to unevenness of the road. A system of equalizing levers so distributes the weight that the wheels are always kept in contact with the rails, no matter how uneven the track.

Motion is transmitted to the driving wheels by means of the same reciprocating parts as on any other engine; the cylinders, which are usually two in number, are cast separate, each with one-half of the *saddle* which carries the front end of the boiler, and to which the frame is attached.

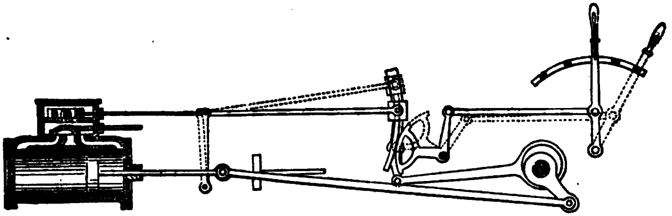


FIG. 1.847—Type of variable cut off generally used by Baldwin in 1853. It consisted of the familiar Gonzenback valve in a separate chest and worked by an independent eccentric and rock shaft. The upper area of the rock shaft was curved so as to form a radius arm on which a sliding block, forming the termination of the upper valve rod, could be adjusted and held at varying distances from the axis, thus producing a variable travel of the upper valve. This device did not give a perfect cut off, as it was not operative in backward gear, but when running forward it would cut off with great accuracy at any point of the stroke. It was quick in its movement, and economical in the consumption of fuel. After a short experience with this arrangement, the partition plate was omitted and the riding cut off substituted.

The boiler is attached directly to the frame at the fire box, there being intermediate supports between the frame and the boiler, their number depending upon the length and type of the locomotive, thus making practically one piece of boiler, cylinders, and frame.

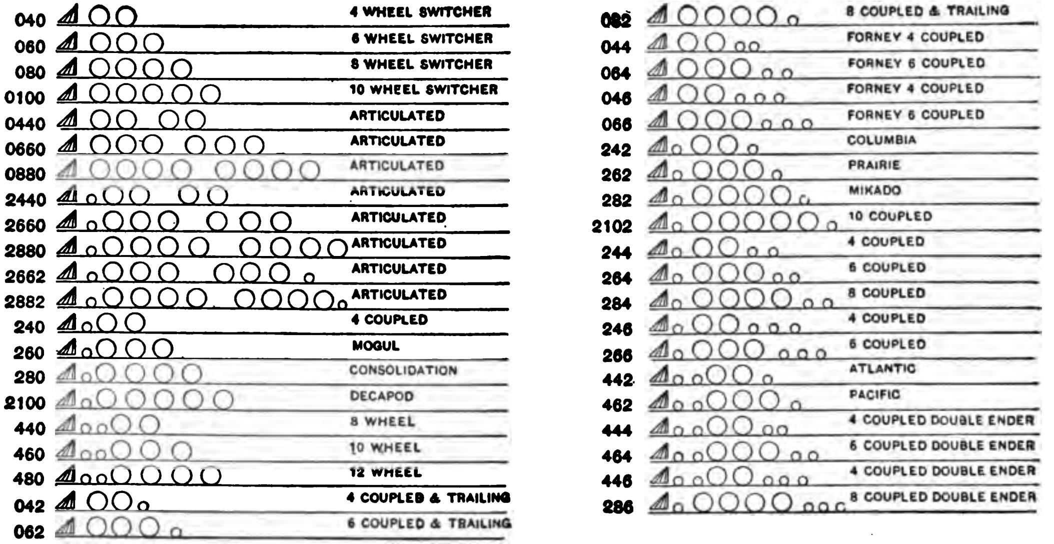
In locomotives having more than one set of driving wheels, the different pairs are connected by means of driving rods, so that all may receive their due share of the rotary motion, induced by the connecting rods.

Besides the driving wheels, there is usually a four wheel truck pivoted to the forward end of the frame, which permits the wheels to follow the track around curves of short radii.

Steam is transmitted to the cylinders by means of a pipe which runs along in the steam space of the boiler to the smoke box, where it branches off to each side to the cylinders.

Classification of Locomotives

[WHYTE'S SYSTEM]



FIGS. 1,848 to 1,888.—Classification of Locomotives. In the above method of classifying locomotives, the different types are represented by the symbols 0—4—0, 0—6—0, 0—8—0, etc. Thus, the engine in fig. 1,848, is classified by the symbol 0—4—0, the first figure 0, being the number of wheels in the truck, the second figure 4, the driving wheels, and the third figure 0, the trailers. Engines of the Mallet system, with two locomotive engines under one boiler, are classified 0—8—8—0c, 2—6—6—2, etc.

The throttle valve is inside the boiler, and is operated by means of a rod passing through the stuffing box in the boiler head and connected with a lever in the cab.

The exhaust steam from the cylinders passes through exhaust nozzles up the stack, thus producing a blast which maintains a strong draft notwithstanding the low stack.

A brake is located between the driving wheels. In starting, the wheels sometimes slip, especially if the track be wet; to prevent this a sand box is arranged on top, from which sand is fed to the track in front of the drivers, by means of a sand valve operated by a rod and lever in the cab.

A so called cow catcher, consisting of numerous wood or metal members radiating from a transverse bar at the front end of the frame to a V shaped section just above the rails, serves to clear the track of any obstruction, such as cattle, etc.

Classification.—The varied conditions of railroad service give rise to numerous types of locomotives; they may be classified in two general divisions: 1, with respect to the steam features, and 2, the running gear or wheel arrangement. Under the first division, locomotives are classed as:

1. Simple,
2. Compound,

according as the steam is expanded in one or two stages.

The second division, which is based on the kind of service, as passenger, freight, switching, etc., naturally includes all types.

The constant increase in the length of trains, speed, etc., has given rise to numerous types of locomotives, as tabulated in figs. 1,848 to 1,888.

The classification here given is that adopted by the American Locomotive Company and is based on the representation by numerals of the number and arrangement of the wheels commencing at the front. Thus, a Mogul locomotive with a two wheel leading truck, six driving wheels and no trailing truck, would be classified as a 260 type.

The total weight is expressed in *pounds* ÷ 1,000. For example, an Atlantic locomotive weighing 176,000 pounds would be classified as a 442-176 type. If the engine be compound, the letter C, should be substituted for the dash; thus, 442 C 176 type. If tanks be used in place of a separate tender, the letter T, should be used in place of the dash. Thus a

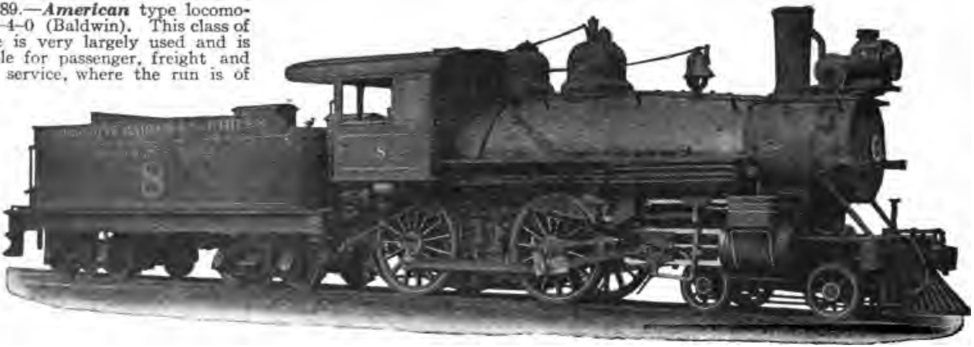


FIG. 1,889.—***American*** type locomotive 4–4–0 (Baldwin). This class of engine is very largely used and is suitable for passenger, freight and mixed service, where the run is of such length as to require a separate tender, or for short lines intended ultimately to be extended. The name "American" was given for the reason that for many years locomotives of this type were used universally for nearly every variety of service throughout the U. S.

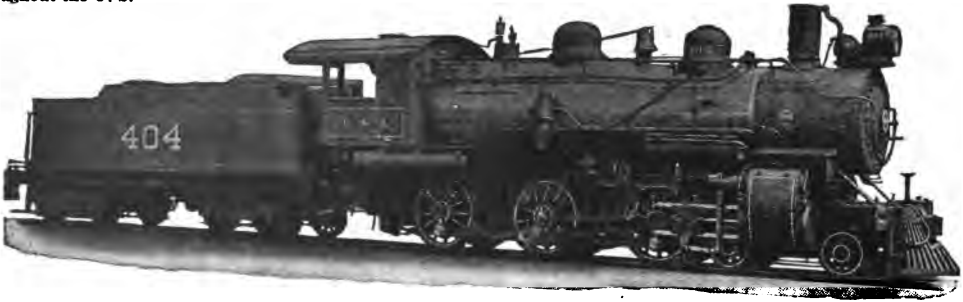


FIG. 1,890.—*Ten wheeled* type locomotive 4–6–0 (Baldwin); suitable where a locomotive of the American type would not afford sufficient power, or where the requisite weight, if carried on only two pairs of driving wheels, would be greater than the rails could safely bear. The greater length of these locomotives admits of a longer boiler, and consequently, greater heating surface. The front and rear driving wheels are, preferably, flanged, and the truck made with swinging bolster.

double end suburban locomotive with two wheeled leading truck, six drivers and six wheeled rear truck, weighing 214,000 pounds, would be a 266 T 214 type.

When locomotive builders started work, they used as few wheels as possible. The first engines had one pair of drivers to make the engine move, and one pair of carrying wheels to keep it on the track. The boilers were built as small as possible to do the work, and were placed between the wheels. Only about one-half of the weight was available for adhesion.

Mr. Baldwin's first engine, the ***Old Ironsides*** (fig. 1,845), had the fire box placed back of the driving axle. In subsequent engines, however, the fire box was located ahead of the driving axle. Mr. Norris, in order to obtain more adhesion, placed the fire box so it would overhang back of the driving axle, which proved of decided advantage in locomotives with a single pair of driving wheels by increasing the proportion of weight on the drivers, and so, by obtaining increased adhesion, it was possible to haul more cars without increasing the total weight of the engine.

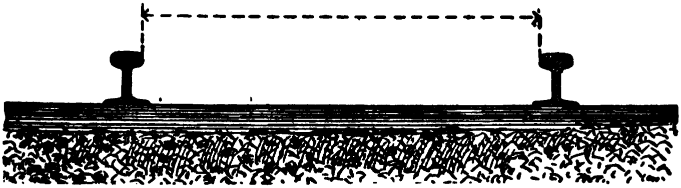


FIG. 1,891.—*Gauge of track.* This term means the distance between the inside edges of the heads of the rails, as indicated in the cut. The distance over the wheel flanges represent the gauge less the amount of play or clearance between the flange and rail. When deciding upon the gauge for a contemplated road, the following suggestions will be found helpful: If the line is to connect with any standard gauge road, the track should correspond and be of the standard broad gauge, which is four feet eight and one-half inches. If such connection be unlikely and narrow gauge is considered preferable, the standard narrow gauge should be adopted, which is three feet. The advantage of adopting one of these standard gauges is that, should it be desirable at any time to sell the equipment, a ready market can be found. For logging railroads the standard gauge of 4 ft. 8½ ins. is generally preferable, as the cars then can have long bolsters and can be heavily loaded without piling the logs too high. While some roads use the same gauge in curves as on tangents it is desirable in order to insure easy riding and reduce wear to widen the gauge in the curves. According to Trautwine, the gauge is usually widened by from ⅛ to ⅜ ins. for each degree of curvature, the maximum amount seldom exceeding one inch.

An important step was the introduction of the ***four-wheel truck,*** which enabled the engine to curve better and be easier on the track.

Later on, to meet the need of greater adhesion, an additional pair of driving wheels was placed behind the fire box and the length of the fire box was increased as much as possible without unduly lengthening the wheel base. This resulted in the ***American type locomotive,*** fig. 1,889.

Another constructive engineer conceived the idea of increasing the number of driving wheels to add to the proportion of weight available for adhesion, and moved the truck forward sufficiently to make room for another pair of driving wheels as shown in fig. 1,890.

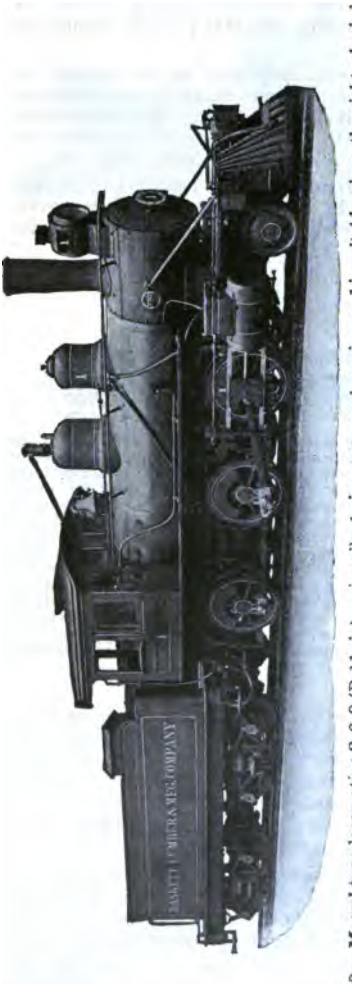


FIG. 1,892.—***Mogul*** type locomotive 2-6-0 (Baldwin); primarily designed for road service, and is suitable where the eight wheeled or American type would not afford sufficient power, or where the requisite weight on the driving wheels, if carried on only two pairs would be greater than the rails could safely bear. The front and rear driving wheels are flanged; the middle pair has no flanges. The pony truck has a swinging bolster and radius bar. ***The design comprises:*** 1, A deep fire box between the middle and rear driving axles. This has the advantage of giving ample depth of fire box, but necessitates a greater spread of wheels than is admissible in some instances; 2, a fire box placed above the frames and over the rear axle. This design admits of the driving wheels being grouped closely together. It answers well where coal is the fuel, but where wood is burned a deep fire box is desirable; 3, driving wheels grouped closely together and a fire box placed entirely back of them. The depth of fire box is sufficient for burning either wood or coal. The short driving wheel base admits of traversing curves of short radius. Connection to the tender is made by means of a radial draw bar passing through the ash pan.

Another one concluded that it was unnecessary to carry so much dead weight on the truck and proposed substituting a two wheeled truck placed forward of the cylinders for the four-wheeled truck previously used. By this means the proportion of weight on the driving wheels was increased from about seventy per cent to eighty five per cent, or even more. The first ***Mogul*** locomotive was thus developed from a Louisville and Nashville ten wheeler. In subsequent locomotives of this type the driving wheels were grouped closer to the cylinders, in order to carry a greater proportion of the weight as shown in fig. 1,892.

The next step in the development of the heavy freight locomotive was the addition of a fourth pair of driving wheels to this type, thus forming the ***Consolidation*** type, fig. 1,893. Previous to this, eight coupled locomotives, with all the weight on the driving wheels, had been in service. In this type Mr. Baldwin placed the first and second pairs of drivers in a

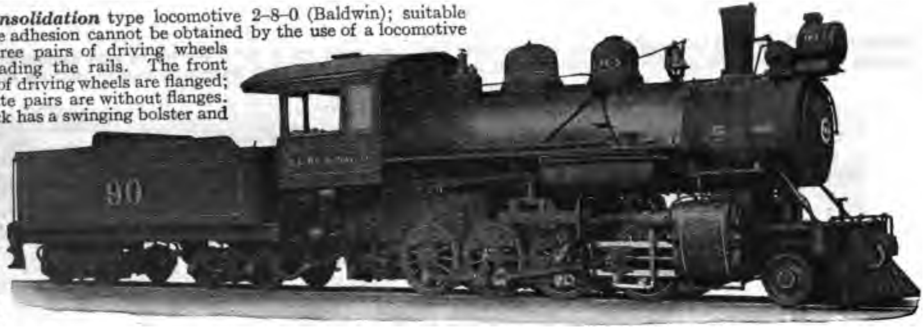


FIG. 1,893.—***Consolidation*** type locomotive 2-8-0 (Baldwin); suitable where adequate adhesion cannot be obtained by the use of a locomotive having only three pairs of driving wheels without overloading the rails. The front and rear pairs of driving wheels are flanged; the intermediate pairs are without flanges. The pony truck has a swinging bolster and radius bar. Ordinarily in this type a long fire box is placed over the rear driving axle, and is especially adapted for burning coal. In some instances such engines have been satisfactorily used for burning wood. For narrow gauge locomotives a deep fire box overhangs the rear driving wheels. The driving wheel base is shorter than in engines of the first mentioned type, because the wheels are placed as close together as possible under the waist of the boiler. In this design there is ample depth between the tubes and the grate for the combustion of wood, while the same plan answers equally well for bituminous coal. The heaviest classes of standard gauge engines are preferably built with the grate placed above the rear pair of driving wheels. This plan provides sufficient grate area, without using a furnace of excessive length.

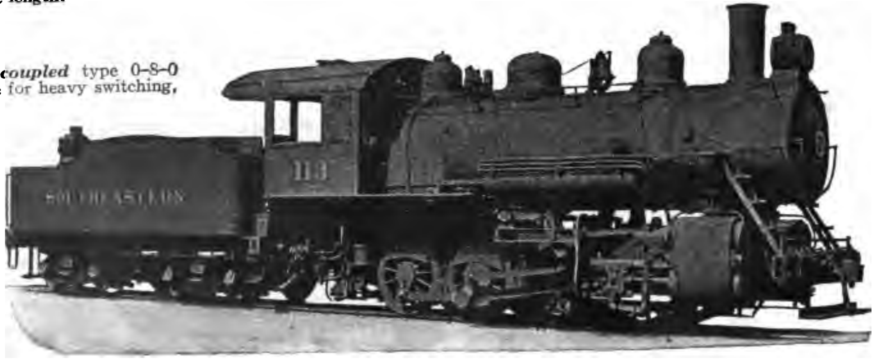


FIG. 1,894.—***Eight coupled*** type 0-8-0 (Baldwin); suitable for heavy switching, where a six coupled locomotive would not afford sufficient power, or where the necessary weight if carried on only three pairs of driving wheels, would be greater than the rails could safely bear. Separate tenders are preferably used with these engines.

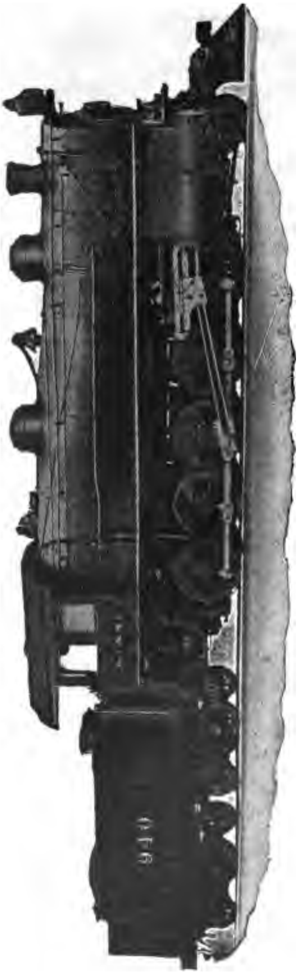


FIG. 1,895.—***Decapod*** type locomotive 2-10-0 (Baldwin); suitable for heavy freight service where an eight coupled locomotive would not afford sufficient power, or where the necessary weight, if carried on four pairs of driving wheels would be greater than the rails could safely bear.

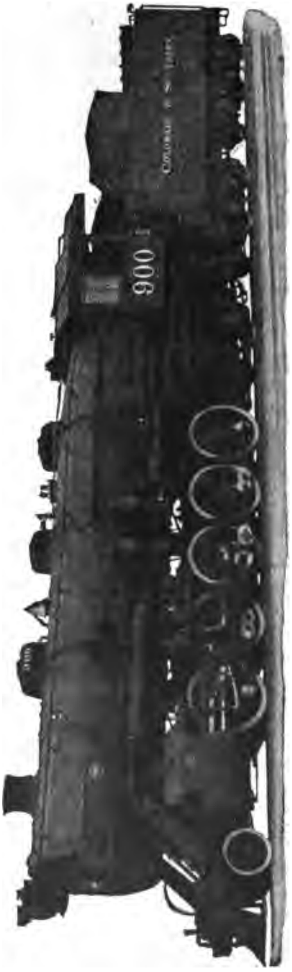


FIG. 1,896.—***Santa Fe*** type locomotive 2-10-2 (Baldwin); a development of the Decapod type, by the addition of a two wheeled rear truck. This feature adds to the flexibility of the wheel base, and improves the curving qualities, especially when running backward.

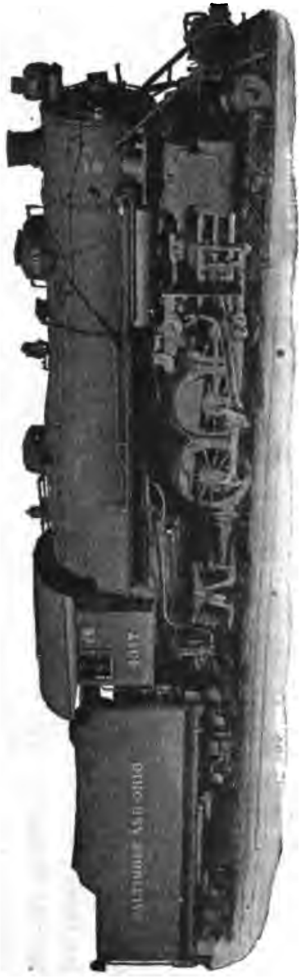


FIG. 1,897.—***Mikado*** type locomotive; developed from the *Consolidation* type, by the addition of a two wheeled rear truck. Where it is frequently necessary to run backward around sharp curves, this feature is of special advantage as it adds to the flexibility of the wheel base and reduces flange wear on the driving wheels. It also permits the use of a deep and wide fire box placed over the trailing wheels.

flexible beam truck, which permitted the locomotives to traverse sharper curves than was possible where the drivers were fixed in a rigid wheel base. This arrangement necessitated placing the cylinders on an angle in the sides of the smoke box. An idea was then advanced to design freight locomotives with five pairs of drivers coupled together (fig. 1,895). This type is known as the ***Decapod.*** In operation, it was found that engines of this class could be run forward around curves without trouble, but in backward motion the rear driving wheels, instead of following the curve, would often turn the rail. To remedy this, a two wheeled trailing truck was introduced, giving rise to the ***Santa Fe type,*** (fig. 1,896), and the ***Mikado type*** (fig. 1,897).

Anatole Mallet, a Frenchman, who for years had endeavored to bring out a successful compound locomotive, built, in 1888, an ***Articulated engine,*** later developed as shown in fig. 1,898.

"For many years the passenger service of the American roads was mainly handled by the American type locomotive (fig. 1,889). The ***Atlantic type,*** (fig. 1,899), was afterwards developed in order to provide a large fire box, and the ***Pacific type,*** (fig. 1,901), have also been introduced for heavy passenger service."

A modified form of the Pacific type is the ***Prairie type*** (fig. 1,900), which has only one pair of leading wheels, so as to bring a greater proportion of the weight on the drivers and thus increase the adhesion.

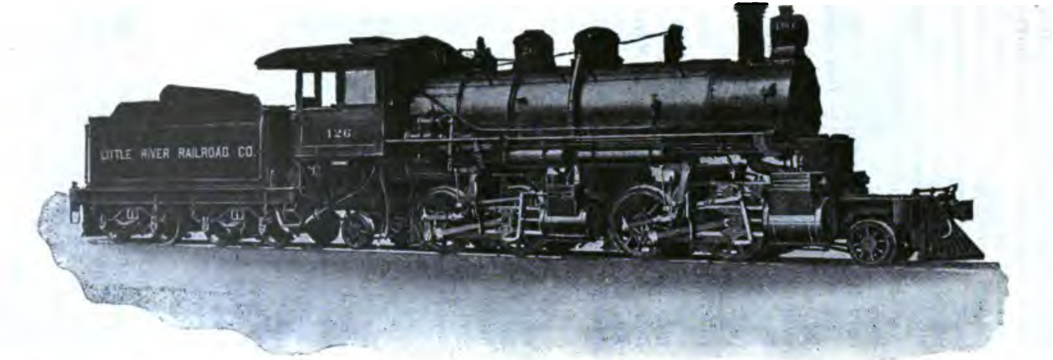


FIG. 1,898.—Mallet articulated type locomotive 2-4-4-2 (Baldwin). It has compound cylinders with two groups of driving wheels. The rear group is driven by the *h.p.* cylinders, and the forward group by the *l.p.* cylinders. The rear frames are in rigid alignment with the boiler, while the front frames can swing about a center pin located on the center line of the engine between the high pressure cylinders. The forward group of wheels thus constitutes a truck giving sufficient flexibility to the wheel base to enable the engine to traverse sharp curves with ease. The receiver pipe between the *high*, and *l.p.* cylinders and the exhaust pipe from the *l.p.* cylinders to the smoke box, are necessarily provided with flexible joints, but as these pipes carry only low pressure steam, practically no difficulty is experienced in keeping the joints tight. The majority of the engines of the the Mallet type are equipped with leading and trailing trucks, which improve the curving qualities and protect the driving wheels against excessive flange wear. The largest engines of this type are preferably built with a separate boiler having a feed water heater in the front section. The separable joint is placed immediately in front of the high pressure cylinders and surrounds an intermediate combustion chamber. The feed water heater is simply a section of the boiler having a tube sheet at each end, and traversed by fire tubes. It is kept constantly filled with water by the injectors, and the overflow from the heater is forced into the boiler proper through check valves, in the usual manner. A superheater or reheater, or both, can readily be installed in a boiler of this type. A number of engines have been built with a reheater, arranged like a Baldwin superheater, located in the smoke box. In some instances the reheater consists of a pipe or group of small tubes, passing through a large flue which traverses the water heater. With this plan the pipe connecting the high and low pressure cylinders is kept as far as possible within the boiler, where it is not exposed to cooling influences.

The Boiler.—The severe duty required of a locomotive boiler determines, in a measure, its general features. The size is limited in that it has to fit in between the frames; notwithstanding this, the boiler must furnish a large quantity of steam at a high pressure

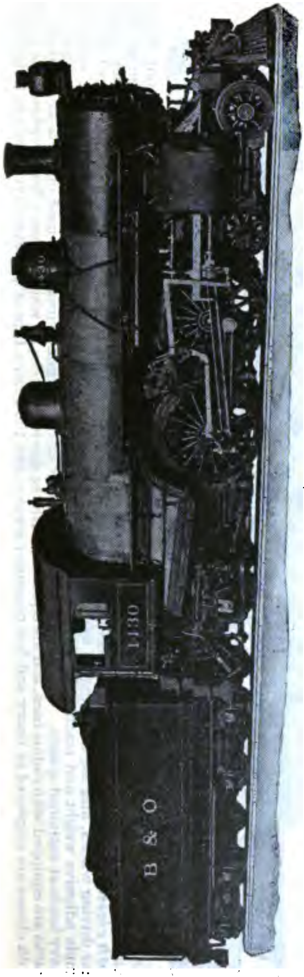
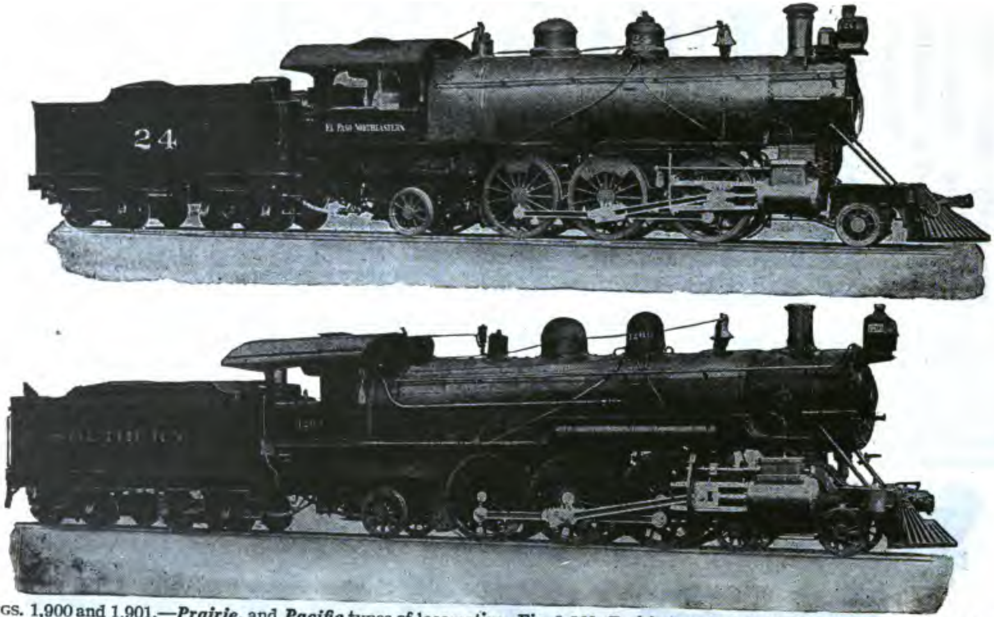


FIG. 1,899.—***Atlantic*** type locomotive 4-4-2 (Baldwin); suitable for high speed passenger service. The driving wheels are located under the waist of the boiler, and the front end of the engine is carried on a four wheeled truck. A fire box having ample grate area and volume is placed back of the rear driving axle, and the overhanging weight is carried by a pair of trailing wheels. This arrangement provides a boiler having great steaming capacity in proportion to the adhesion—an essential feature of a high speed locomotive. In locomotives of this type the fire box may be placed entirely back of the driving wheels if desired, thus allowing an increased width of furnace. It is preferable to use a rigid pair of trailing wheels, with both pairs of driving wheels flanged. The leading truck is provided with a swing bolster, and the relation of the rigid to the total wheel base is practically the same as in a locomotive of the ten wheeled type. The compact grouping of the driving wheels permits the use of short coupling rods, thus reducing the liability of breakage when running at high speed.

in order that, with comparatively small cylinders, the locomotive may develop considerable power. These conditions make it necessary to sacrifice economy for large capacity, hence intense combustion is necessary under forced draught; in other words, a large quantity of coal must be burned on a fire grate of limited area. To utilize advantageously the heat thus generated, a large heating surface is necessary. This is provided by passing the products of combustion through very many small tubes.

Forced draught is obtained by discharging the exhaust steam from the cylinders into the smoke stack. The exhaust pipes connect directly beneath the stack with a vertical *blast pipe*, which has at its upper end an exhaust *nozzle*. The latter is



FIGS. 1,900 and 1,901.—***Prairie,*** and ***Pacific*** types of locomotive. Fig. 1,900, Prairie type 2–6–2 (Baldwin); fig. 1,901, Pacific type 4–6–2 (Baldwin). Locomotives of the Prairie and Pacific types possess a number of features in common, the principal difference being that the Prairie type has a two wheeled leading truck, while in the Pacific type the leading truck has four wheels. ***In both classes*** the three pairs of driving wheels are grouped under the waist of the boiler. The fire box is placed back of the rear driving wheels, and the overhanging weight is supported by a two wheeled rear truck. The result is a locomotive having ample adhesive weight and tractive force, together with a boiler of high steaming capacity, thus enabling the engine to haul heavy loads at sustained speeds. Trailing trucks, with either inside or outside journals, are used on these locomotives. These trucks are equipped with radius bars, and are equalized with the driving wheels; by which means a flexible wheel base is obtained. Both classes are employed in heavy and fast passenger service, and the Prairie type is also extensively used in freight service.

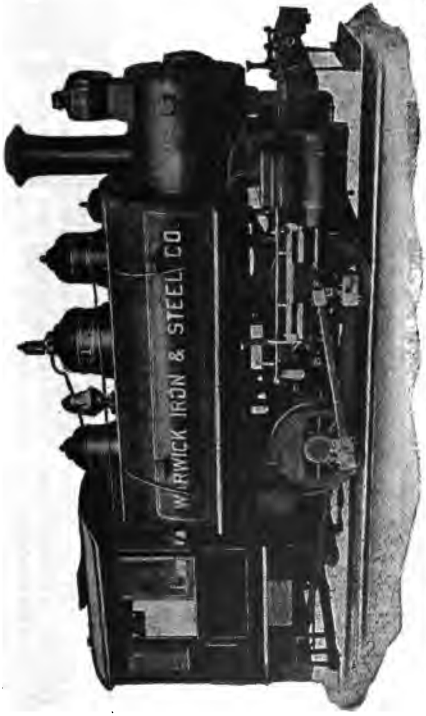


FIG. 1,902.—**Four coupled** switching locomotive 0–4–0 (Baldwin); **the simplest form of locomotive** construction. All the **weight is on the driving wheels** and utilized for adhesion. The sharpest curves can be passed without difficulty on account of **the short wheel base. Curves** of fifty feet radius may be easily traversed by the smaller classes, while seventy-five to ninety **feet radius can be set down as** a minimum for the larger classes. Engines of this type can be run equally well in either direction. **When the run is short, a** sufficient supply of fuel and water can be carried on the engine. For longer runs, where a larger **amount of fuel and water is** required, a separate tender is supplied. If desired, the tender tank is made with a sloping back. **A separate tender is also an ad**vantage on exceptionally narrow track, as admitting of a lower center of gravity than if the tank **were placed on the boiler.**

sometimes made adjustable so that its area may be varied to regulate the draught. In some cases there are separate blast pipes for each cylinder.

A standard form of locomotive boiler is shown in fig. 1,903. It includes three principal parts: 1, a *fire box* A, which is surrounded by water, 2, a *shell* B, attached to the fire box at one end, and to 3, a *smoke box* at the other end.

A great number of tubes connect the fire box and smoke box, through

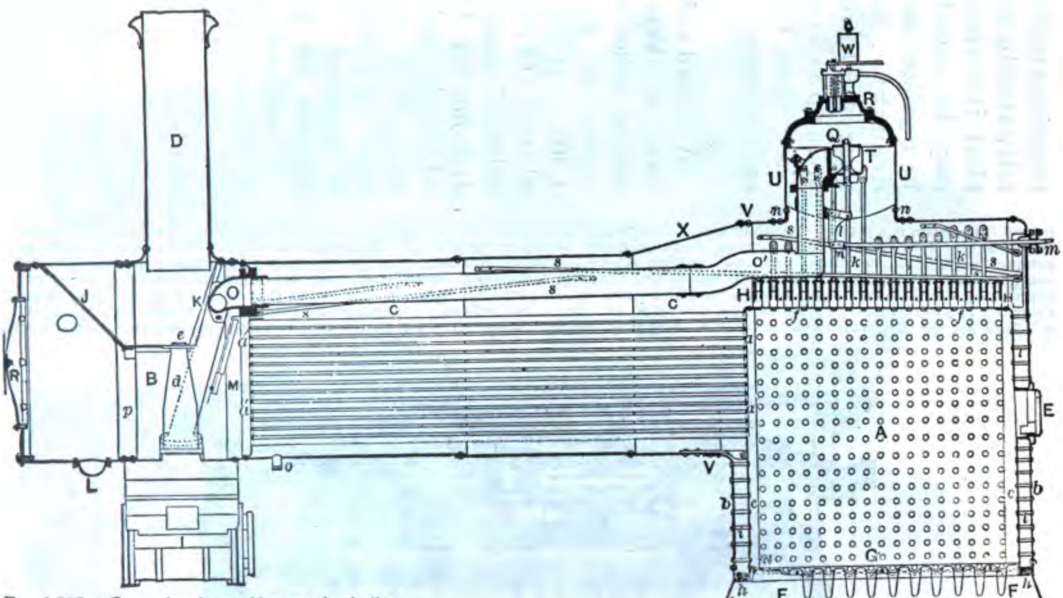


FIG. 1,903.—Crown bar form of locomotive boiler.

which the products of combustion pass from the fire box to the smoke box. These tubes being of considerable length and surrounded by water present a large heating surface. The fire box is made up of inner and outer plates, with the intervening space filled with water. G, is the grate and E, the furnace door. An *ash pan* FF, is placed below the grate and is provided with suitable dampers. The upper inner plate of the fire box is called the *crown sheet*, and the forward inner plate the *tube sheet*. A second tube sheet forms a partition between the boiler proper and the smoke box.

The tubes, which are ordinarily two inches in diameter, are expanded into holes drilled in the two sheets.

Above the fire box is a vertical cylindrical chamber called the *steam dome*, having attached on top, the safety valve and whistle W. Inside is the throttle valve T, attached to an elbow on the main steam pipe. The latter runs forward through the forward tube sheet, and branches downward to the cylinders.

The exhaust pipe *d*, and the nozzle *e*, are seen directly below the *smoke stack* D.

In order to prevent unequal distribution in the flow of the products of combustion through the tubes, a *baffle plate* M, is placed so as to diminish the draught through the upper tubes. A wire netting or *spark arrester* J, is provided, through which the smoke and gases must pass before entering the smoke stack.

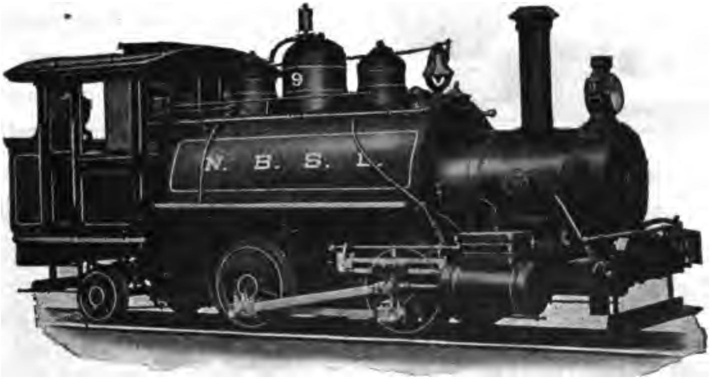


Fig. 1,904.—Four coupled with two wheel rear truck type locomotive 0–4–2 (Baldwin); particularly serviceable for operating short lines, where limited water and fuel capacity will answer. These locomotives have their driving wheels equalized together, the truck being center bearing, with swinging bolster and radius bar. Having a comparatively long total wheel base and a short rigid wheel base, they are steady, and ride smoothly, without plunging; curve readily, and cause little wear of track. The fuel is carried on the engine frames at the back; the water is carried either in a saddle tank on the boiler, in side tanks on each side of the boiler, or in a tank back of the cab. The latter plan is better for light rails. If the tank be placed on the boiler, its weight adds to the adhesion and increases the hauling capacity, greater space is afforded the engine men in the cab, and a larger supply of fuel may be carried. The weight is well distributed, the principal portion being carried on equalizing levers between the driving wheels, thus affording an equal distribution on the driving wheels. The pony truck carries the weight of the fuel or the fuel and water, as the case may be, with a part of the weight of the overhanging fire box. These locomotives are well adapted for running in either direction without turning.

At the end of the smoke box is a door R, giving access to the interior for cleaning. Any accumulation of cinders is swept through the opening P.

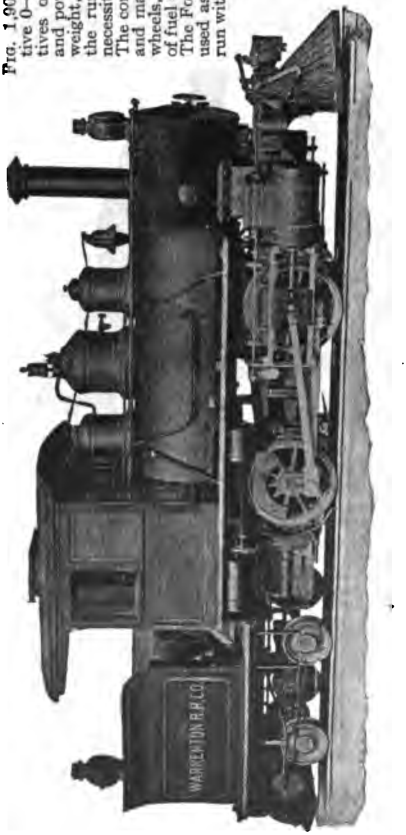


FIG. 1,905.—Forney type locomotive 0-4-4 (Baldwin). Locomotives of this type are compact and powerful for their aggregate weight, and are suitable where the run is not long enough to necessitate a separate tender. The constant weight of the boiler and machinery is on the driving wheels, while the variable weight of fuel and water is on the truck. The Forney type locomotives are used as "double enders," being run with equal facility forward or backward. The driving wheels are equalized together; the truck is center-bearing and has a swinging bolster. These locomotives readily traverse curves of short radius. The fuel and water are carried at the rear of the cab.

The figure shows the cylinders and saddle to which the forward end of the boiler is attached.

The problem of adapting the locomotive boiler to different kinds and grades of coal, and of overcoming structural defects, has caused changes in boiler design from time to time, altering considerably, in some cases, the form and center of gravity. Boilers may be classified

1. According to their shape, as:

   a Straight top;
   b Wagon top;
   c Extended wagon top;
   d Turtle back.

2. With respect to the type of fire box, as:

   a Narrow fire box;

b. Wide fire box;

c. Belpaire;

d. Wootten;

e. Corrugated furnace.

The ***straight top*** boiler has a shell of uniform diameter, with its top, level with the top of the outside plate of the fire box.

The ***wagon top*** boiler has a steam dome over the fire box and a conical or sloping course of plates next to the fire box and tapering down to the cylindrical courses.

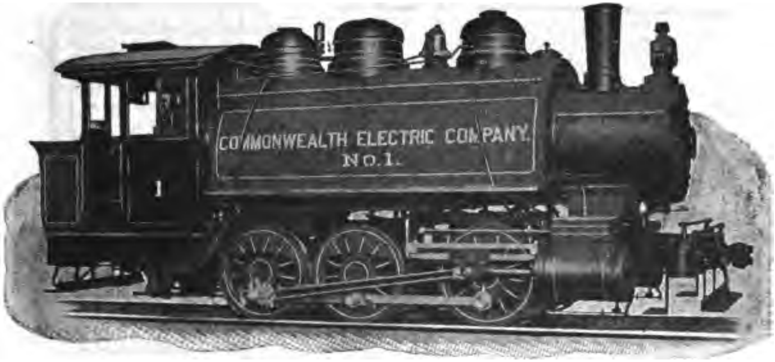


FIG. 1,906.—Six coupled switching 0–6–0 (Baldwin); suitable where the conditions are such as to make it advisable to distribute the weight over more than two pairs of driving wheels. Where the run is short a tender is unnecessary and the tank can be placed on the top or at the sides of the boiler. Under other conditions the separate tender is more convenient, as it affords a greater supply of fuel and water than the tank locomotive, and longer runs can be made. In the heavier classes for narrow gauge the separate tender is preferable, as it avoids raising the center of gravity of the locomotive.

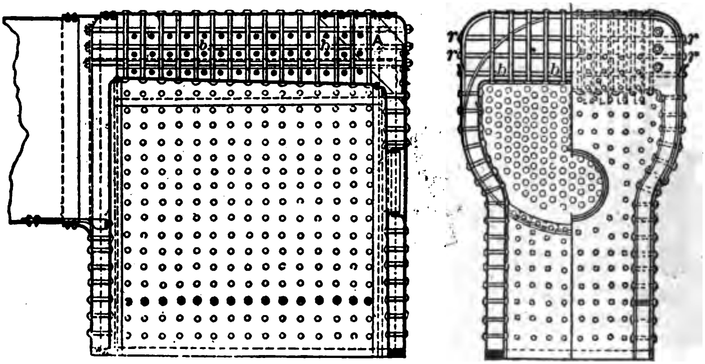
The ***extended wagon top*** boiler has a shell made up of one or more cylindrical courses of reduced diameter, adjacent to the smoke box. The object of the cylindrical course next to the fire box is to provide a place for the steam dome in front of the fire box, and thus do away with a crown bar staying over the crown sheet as is necessary in the wagon top boiler.

The ***turtle back*** boiler is a name sometimes applied to the early form of the Wootten boiler, on account of the sloping sheet above the crown sheet.

A ***narrow fire box*** boiler has a deep fire box narrow enough to extend down between the frames; this type is suited to bituminous coal, or anthracite of good quality. The restricted grate area precludes the use of coal of low grade, except for light service.

A ***wide fire box*** boiler has a wide shallow fire box resting on the frames and extending out beyond them at the sides. The increased grate area thus obtained adapts the boiler to heavy service, or to the burning of poorer grades of coal than is permissible with the narrow fire box. The wide fire box boiler is in general use on locomotives of the Prairie, Pacific, Mikado and Santa Fe types.

A ***Belpaire boiler*** has a fire box with a flat crown sheet joining the side sheets by a curve of short radius, and having the outside crown sheet and the upper part of the outer side sheets flat and parallel to those of the inner fire box. These flat parallel plates are stayed with direct vertical, and transverse horizontal stays, obviating the necessity of crown bars to support and strengthen the crown sheet.



Figs. 1,907 and 1,908.—Longitudinal and cross sections of Belpaire fire box showing crown sheet and sides stayed with parallel stay bolts.

A ***Wootten boiler*** has a fire box very wide and shallow, and a curved crown sheet of large radius, being designed to burn a poor grade of coal or fine culm from the anthracite region. This type was developed in 1877, many being put in service and continue to be used. On account of the space required for the fire box, the cab is placed forward, with the steam dome in the cab; in some cases the dome is placed forward or back of the cab. Formerly the top sheet over crown was sloped down to the back end. In later construction, this sheet was carried back straight, thus providing a larger water and steam space over the crown sheet, the top sheet from top radius to bend at the sides being flat. This type of boiler provides the most liberal grate area, hence it is suited to burning the poorest grades of coal.

A ***corrugated furnace boiler*** has a fire box consisting of a large cylindrical flue which is channeled or corrugated circumferentially, thus adding to its strength, and providing for expansion and contraction. The furnace is identical with that of the Scotch marine boiler with a somewhat larger diameter; it was first applied to German and Austrian locomotives by Lentz, and later to American locomotives by Vanderbilt. Undue prominence has been given these names, especially the latter, which seemed unwarranted, as the corrugated furnace boiler is not an invention, simply a new application of a type of furnace which was in use many years before being applied to the locomotive. The absence of stays is the chief advantage gained, but the grate being limited to rather small area, necessarily, restricts the boiler to the better grade of coal.

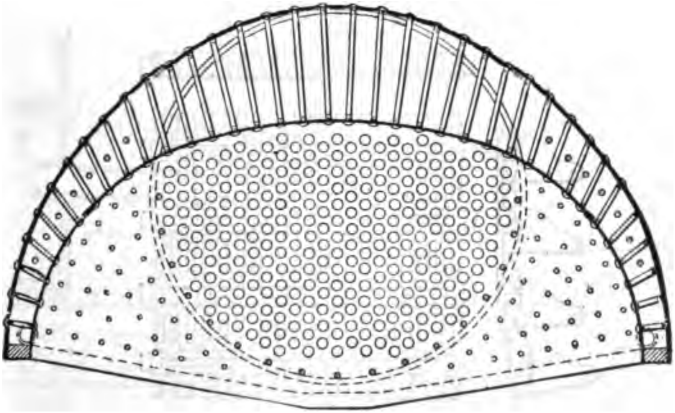
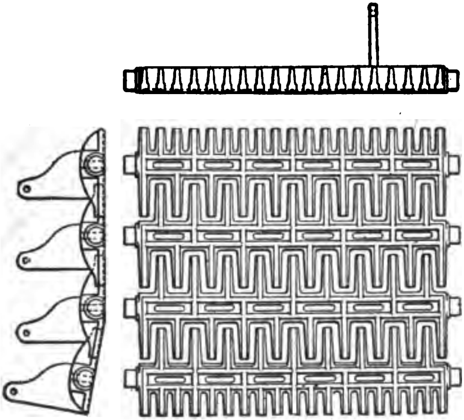


Fig. 1,909.—Cross Section through fire box of Wootten boiler. Where fine anthracite coal is burned, a much larger fire box is required than in the case of bituminous coal. The fire box need not, however, be as deep. The Wootten fire box, as shown, is very wide and very shallow. The boiler is raised so that the grates are above the tops of the driving wheels. The fire box can, therefore, be made as wide as the clearance space of the roadway will permit; the stays are arranged semi-radially.

Grate Area and Heating Surface.—A locomotive boiler differs from a stationary or marine boiler in that it is necessarily of smaller relative proportions for the same duty. This necessitates burning a larger amount of fuel and generating a greater supply of steam in a given time in proportion to its size. To do this, means must be resorted to for increasing the rate of combustion, and steam of high pressure must be carried.

The grate upon which the fuel is burned is placed at the bottom of the fire box. The form of the grate and the dimension of the air spaces vary with the fuel used.

For burning coal movable grates are used, a great variety of forms having been tried, some of which are shown in the accompanying cuts.



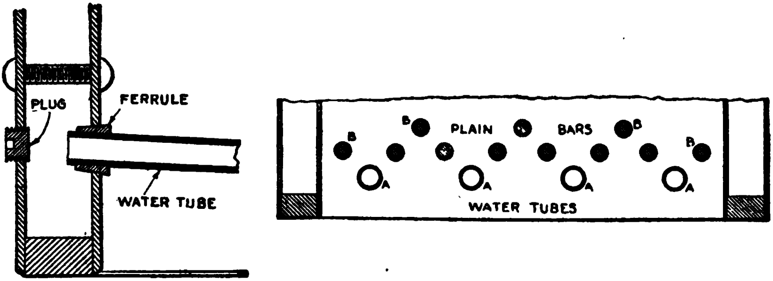
Figs. 1,910 to 1,912.—A type of movable grate for bituminous coal. It consists of a central bar carrying the fingers which interlock with each other, forming air spaces. The grate can be moved by a bar beneath connecting the downwardly projecting arms of the several bars. For dumping the sections are turned to an inclined position. At the front end of the fire box there is usually an independent grate consisting of a perforated flat plate. This can be operated separately from the remainder of the grate. It is used when it is desired to clean the fire without dumping the whole. This plate is opened and the clinkers contained in the bed of coals can be pushed through the opening, thus formed, into the ash pan beneath.

Because of the high rate of combustion necessary, the ratio of heating surface to grate area is higher than for stationary or marine boilers.

In practice about 50 to 75 sq. ft. of heating surface are provided per sq. ft. of grate, the proportion of heating surface to grate area being governed largely by the kind of fuel used and operating conditions.

Anthracite coal and the poorer qualities of fuel require larger grates than good bituminous coal or wood. It is, however, quite certain that the larger the boiler, and the greater its heating surface in proportion to the steam it must generate, other things being equal, the more economical will it be in its consumption of fuel, or, in other words, the more water will it evaporate per pound of coal.

The Smoke Box.—This is a chamber located at the forward end of the boiler into which the gases of combustion enter from the tubes and pass out through the smoke stack placed on top. The smoke box contains the exhaust pipe, exhaust nozzle, netting, diaphragm plate, cinder trap and the super-heater on engines fitted with the latter device.



Figs. 1,913 and 1,914.—Cross section and end view of combination bar and water grate for anthracite coal. The water section of the grate is formed of tubes expanded into ferrules in the back sheets of the fire box. They are inclined to insure circulation of water. The back sheet is tapped for plugs to permit inserting, expanding and cleaning the tubes as well as affording means for removal for replacement when necessary. Grates made entirely of water tubes are rarely used. The water tubes A, fig. 1,914, being spaced some distance apart and between them plain bars B, being passed through tubes expanded into the sheets. By turning or removing the bars, the fire may be shaken or dumped.

In operation the gases strike upon the diaphragm plate G, fig. 1,915 and are deflected downward, and then turn up and pass through the netting F, and out at the stack which is located directly above the exhaust nozzle E. The figure shows the various details of construction.

The Dry Pipe and Throttle Valve.—The dry pipe is provided to carry dry steam to the cylinders, the steam entering the pipe

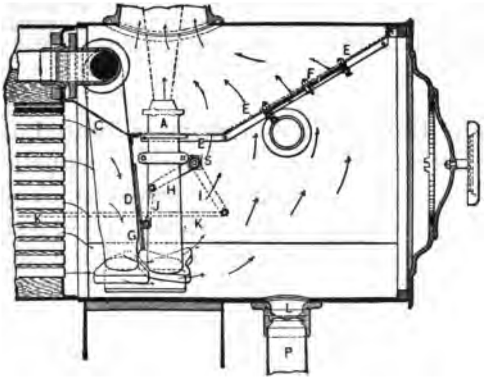
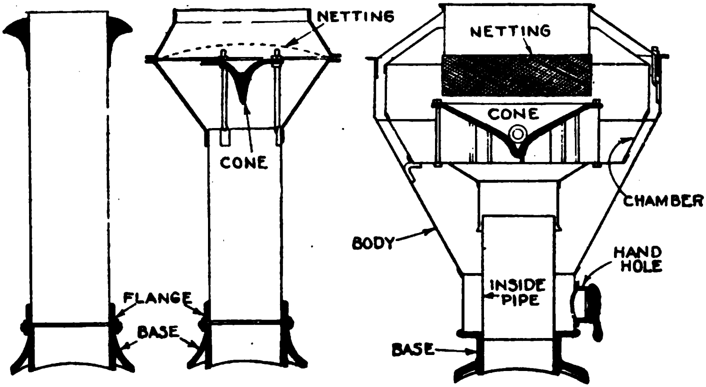


Fig. 1,915.—Interior view of smoke box showing fixtures contained therein. ***The parts are:*** A, exhaust nozzles; B, stack; C and D, deflectors; EE, wire netting; F, manhole; G, draught control door; H and I, sliding door bell crank; JK, bell crank rods; L, slide; P, cleaning pipe.



Figs. 1,916 to 1,918.—Various types of smoke stack. Fig. 1,916, plain cylindrical stack. Fig. 1,917, diamond stack consisting of a central pipe above the axial line of which there is a cast iron cone shaped deflector, against which the sparks and cinders strike, which act causes many of them to fall, besides reducing the force with which the others strike the wire netting that is put over the top of the pipe in order to keep live cinders from escaping. Below the cone there is a chamber into which the sparks may fall and cool. The diamond stack is adapted to bituminous coal and wood, when the smoke box is small. Fig. 1,918, stack for wood; it is a very wide double cone top surrounding a central cylindrical pipe, a cone deflector, and a central wire netting. The space around the central pipe serves as a receptacle for cinders and is supplied with a handhole through which they may be removed. The names of the parts are given in the figures.

at the throttle valve in the dome as shown in fig. 1,919. It will be found that after the throttle is opened the water in the boiler has raised one gauge or more. This is caused by the release of

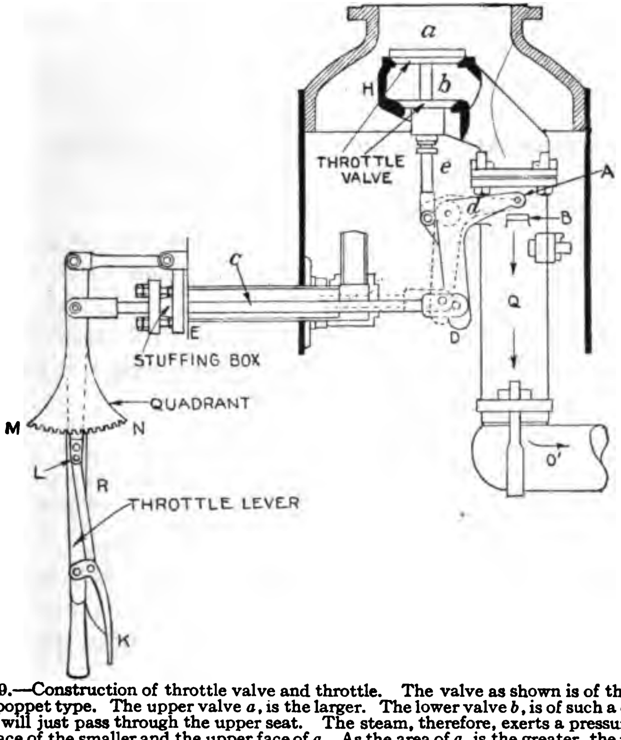


FIG. 1,919.—Construction of throttle valve and throttle. The valve as shown is of the double sealed poppet type. The upper valve *a*, is the larger. The lower valve *b*, is of such a diameter that it will just pass through the upper seat. The steam, therefore, exerts a pressure on the lower face of the smaller and the upper face of *a*. As the area of *a*, is the greater, the resultant tendency is to hold the valve closed. The valve is, therefore, partially balanced. It is, however, usually difficult to open large throttle valves, such as are now used on locomotives with high boiler pressure, and with an ordinary direct form of leverage it is necessary to give a strong, quick jerk to the throttle lever before the valve can be started from its seat. The leverage arrangements as shown obviate this difficulty. The rod *c*, connects with the lever in the cab and communicates its pull to the bell crank *d*, whence it is carried by the stem *e* to the valve. The pivot of the bell crank is provided with a slotted hole. At the start the length of the lever arm is about 2¼ inches while the long arm is 9½ inches. After the valve has been lifted from its seat and is free from the excess of pressure on *a*, the projecting horn A, on the back of the bell crank comes in contact with the bracket B, on the side of the throttle pipe and the crank takes the position shown by the dotted lines in the figure. The end of the horn then becomes the pivot and the length of the short arm of the lever is changed to 9½ inches and of the long arm to about 11½ inches.

pressure on the liberating surface, which in turn causes rapid ebullition or boiling of the water.

The action is such that more or less spray or moisture is projected above the liberating surface, hence, by locating the throttle valve in the top of the steam dome, steam is obtained as near dry as possible.

The Boiler Attachments.—There are a number of devices and fittings that must be used on a locomotive boiler for its proper and safe working. Most of these are common with those on other types of boilers and consist of water gauge, gauge cocks, steam gauge, throttle, injectors, blower, sand valve and pipes, bell, whistle, lights, etc.

In addition the air brake and its control devices are also mounted on the boiler and which are later described in detail.

The "pop" type of safety valve is used.

This valve is so constructed that after opening at the predetermined pressure it blows until the pressure of the steam is reduced, one or more pounds, in order to prevent the continual opening and closing or "vibrating" of the valve when the steam is at the blow off pressure.

Formerly direct connected feed pumps were used, but because of their inaccessible location while in operation and the high rotative speed of locomotives, they have been replaced by injectors.

Two injectors being used, one on each side of the boiler usually in the cab, but in some cases in front of the cab. In locomotives of later design, reflex water gauges are provided in addition to the three gauge cocks for ascertaining the water level in the boiler.

The sand box is usually a cylindrical receptacle which is made of sheet iron with a cast iron base and top.

It is generally placed on top of the boiler and is intended to carry a supply of dry sand, which is scattered on the rails in front of the driving wheels when the latter are liable to slip. This is done by pipes, one on each side of the engine. They lead from the sand box to within a few inches of the rail.

At the upper end and inside the sand box they each have a valve which is operated by a lever connected to the cab by a rod so that the locomotive runner can open or close the valve at pleasure. The sand box has an opening on top through which the sand is supplied to the box. This opening has a loose cover to exclude rain and dirt from the sand.

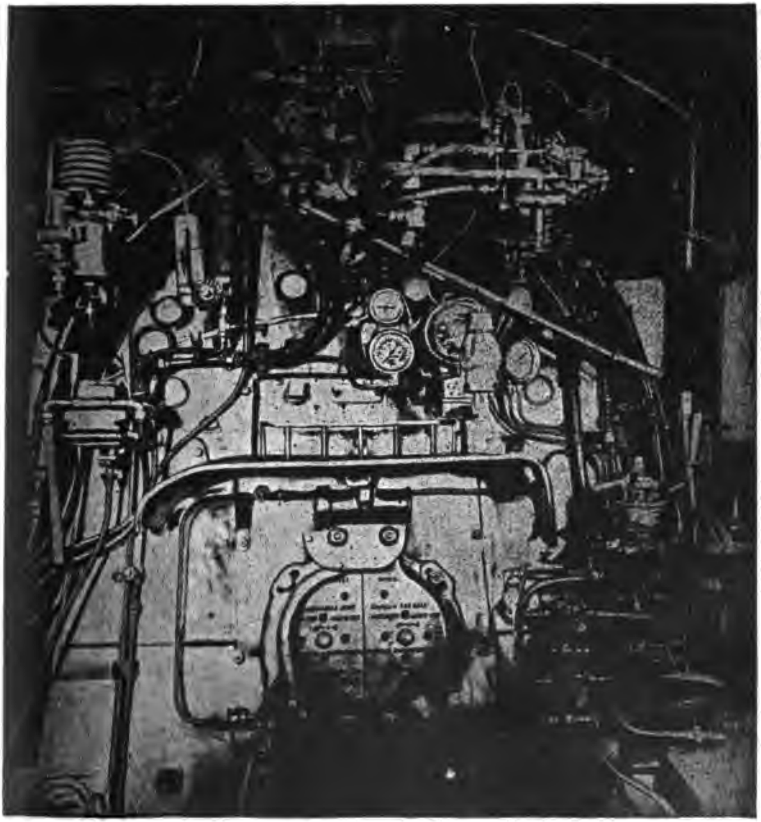
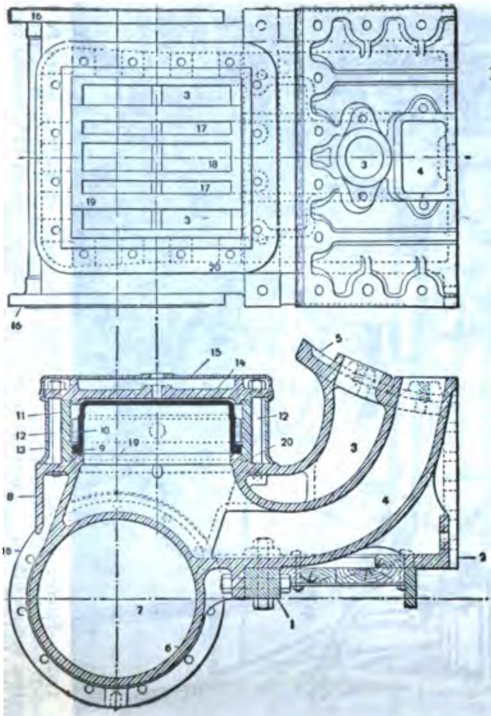
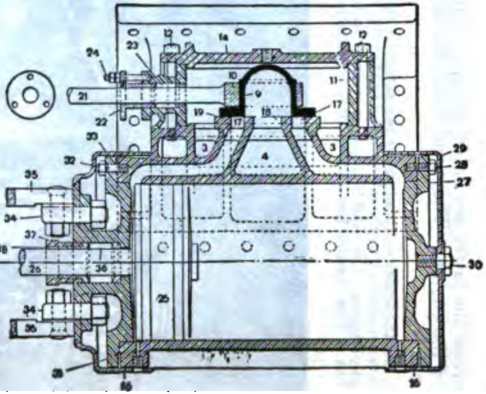


FIG. 1,920.—Interior view of cab showing boiler attachments of Baldwin, ***Santa Fe type*** (4-6-2) locomotive. Cylinders 17½ and 29×28; heating surface 3,475 sq. ft.; superheater heating surface 619 sq. ft.; grate area 58 sq. ft.; diameter drivers 73″; working pressure 210 lbs.; fuel soft coal; wheel base: engine 35′ 1″; tota 70′ 6¼″.





FIGS. 1,921 to 1,923.—Details of cylinder casting. ***The parts are:*** 1, frame; 2, inside face of saddle; 3, steam passage; 4, exhaust passage; 5, smoke box seat; 6, cylinder shell; 7, cylinder bore; 8, casing apron; 9, valve; 10, valve yoke; 11, steam chest; 12, steam chest studs; 13, steam chest casing; 14, steam chest cover; 15, steam chest casing cover; 16, cylinder flanges; 17, steam ports; 18, exhaust ports; 19, valve seat; 20, steam chest seat; 21, valve stem; 22, valve stem gland; 23, valve stem stuffing box; 24, valve stem gland studs; 25, piston; 26, piston rod; 27, front cylinder head; 28, front cylinder head stud; 29, front cylinder head casing; 30, front cylinder head casing stud; 31, back cylinder head; 32, back cylinder head stud; 33, back cylinder head casing; 34, guide brackets; 35, guides; 36, piston rod stuffing box; 37, piston rod gland; 38, piston rod gland bushing.

The Cylinders and Valve Gear.—The design of the cylinders vary with different boilers, but are cast separate and so constructed that they will fit on either side, that is, they are neither rights nor lefts. Each cylinder casting comprises three parts:

1. The cylinder;
2. Steam chest;
3. Half saddle.

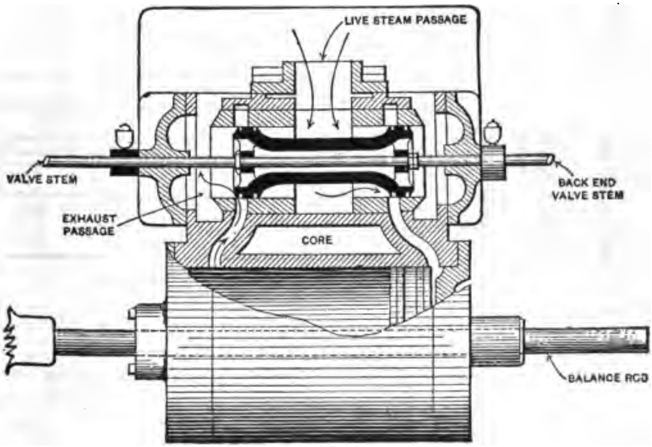


Fig. 1,924.—Sectional view through valve ports and valve chest showing piston valve of the inside admission type.

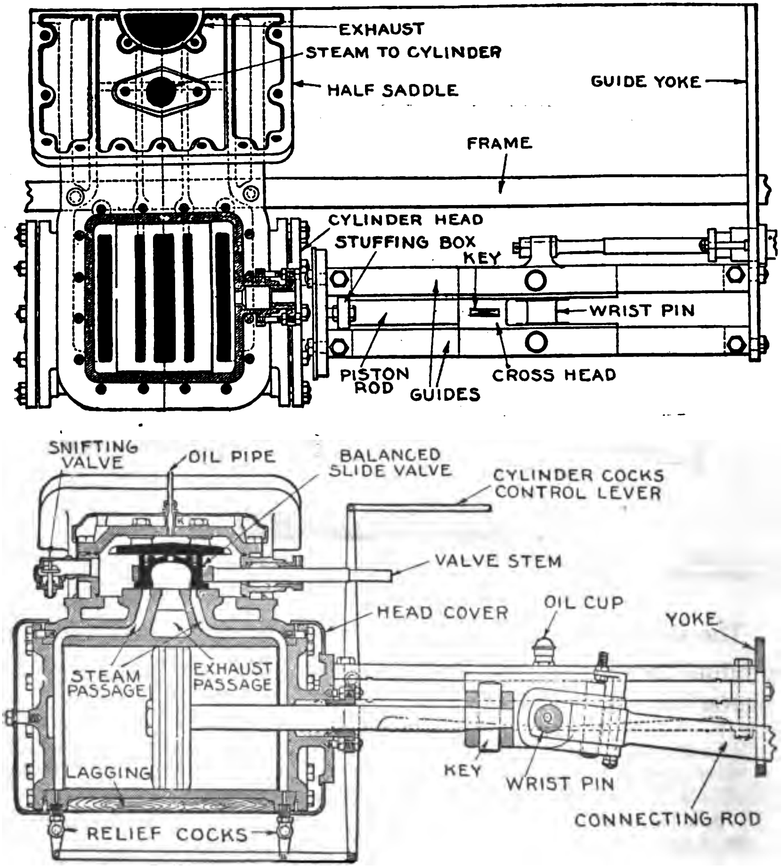
The slide valve has long been the regular type used, but the gradual increase in boiler pressure, has resulted in the use of piston valves.

In the slide valve type, the steam chest is on top of the cylinder and the valve seat in most locomotives is a separate piece bolted to the cylinder casting thus permitting removal when worn out.

Some engines have the seat cast with the cylinder and when worn down have the separate or false seat attached.

Figs. 1,921 to 1,923, show a plan and sectional views of a cylinder casting and method of bolting it to the frame, the latter being seen at 1 in fig. 1,922.

Piston and Piston Rod.—There are numerous types of pistons. One of the most usual designs consists of a spider and a follower plate. The spider consists of a hub and radial arms, the follower plate being bolted to the spider.



Figs. 1,925 and 1,926.—Plan and elevation of cylinder assembly showing cylinder, half saddle, guides, valve, piston, piston rod, connecting rod, guide yoke, etc., with names of parts.

The built up piston is provided with rings which are carried by a T ring. The piston is usually attached to the rod by a key.

The Guides and Cross Head.—Numerous types of guides are used, and they may be divided with respect to the number and construction into several classes, as

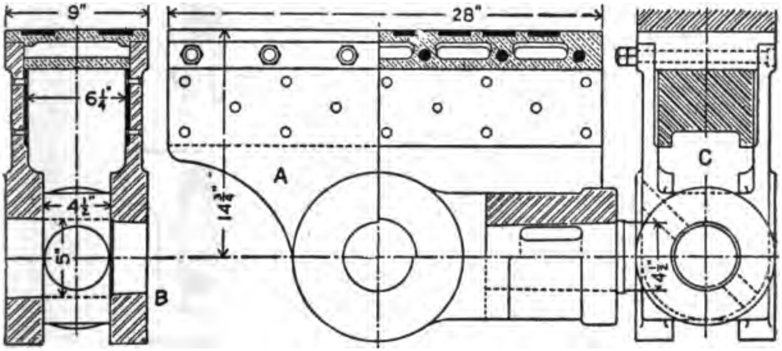


FIG. 1,927.—Slipper type cross head for large freight locomotive. *In construction,* the brass shoe, made hollow to save weight, unites between two bars and is held to the body by tightly fitted through bolts. The flanges which unite the shoe to the body of the cross head are lined with thin brass plates riveted. The main bearing surface on the top of the shoe is composite, shallow pockets are drilled in the brass and filled with babbitt metal.

NOTE.—*Diameter of piston.*—Since the length of stroke is usually fixed to harmonize with the arrangement and diameter of the driving wheels, the determination of the size of the cylinder usually consists in a calculation of the diameter. In order to make the calculation of diameter of the driving wheels, the weight on the driving wheels, the boiler pressure and the stroke of the piston must be known. With this data the diameter of the cylinder can be calculated by the following formula:

$$A = \frac{\frac{W}{4} \times C}{.90\ P \times 48} = \frac{W \times C}{14.4\ P \times S}$$

or

$$A = \frac{W \times D \times 3.1416}{14.4\ P \times S} = \frac{.218\ W \times D}{P \times S}$$

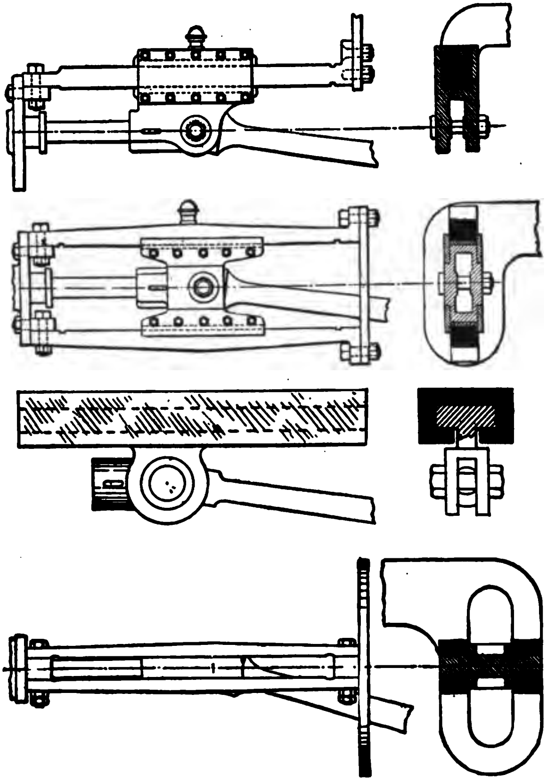
in which A = area of each piston in square inches; W = weight on driving wheels; C = circumference of driving wheels = 3.1416 D; D = diameter of driving wheels in inches; P = boiler pressure in pounds per square inch; S = stroke of piston in inches. *Example.*—Take an engine in which the weight on the driving wheels is 95,000 pounds; the diameter of the driving wheels is 62 inches; the boiler pressure 180 pounds per square inch and the piston stroke is 24 inches. The last formula then becomes:

$$A = \frac{.218 \times 95{,}000 \times 62}{180 \times 24} = 297.2 \text{ sq. in.}$$

or the diameter of the cylinder should be 19.44 inches. Such a cylinder would in practice probably be made 19 inches in diameter.

1. One bar;
2. Two bar;
3. Three bar, in one piece;
4. Four bar.

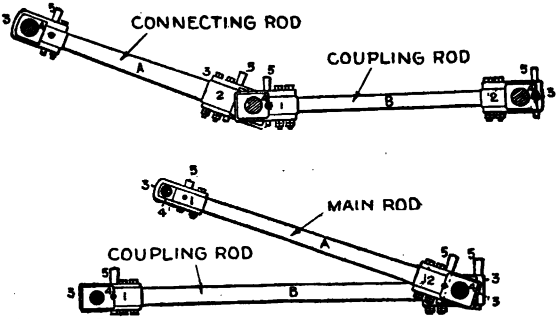
These various types are shown in the accompanying illustrations.



FIGS. 1,928 to 1,935.—Various guides and cross heads. The one bar guide, figs. 1,928 and 1,929 is used on locomotives having small driving wheels to avoid the rubbing surface too near the ground where it would be more exposed to dirt. This type is liable to bend and cause the piston rod to break at the cross head. Figs. 1,930 and 1,931 show a two bar guide, and figs. 1,932 and 1,933, a three bar guide made in one piece. The latter type gives a very large surface for the ahead motion, and is so shaped that it is well protected from dirt. The four bar guide shown in figs. 1,934 and 1,935 was formerly almost universally used.

Connecting and Coupling Rods.—These are made of steel because the reduced weight thus possible reduces the inertia effect which at high speed is considerable and brings severe stresses on the rails and crank pins.

Figs. 1,936 and 1,937 show two arrangements of rods. Formerly provision was made to take up wear in the coupling rods, but later designs as in figs. 1,938 and 1,939 have no means for adjustment.



FIGS. 1,936 and 1,937.—Connecting and coupling rods. Fig. 1,936, outside connected; fig. 1,937 inside connected.

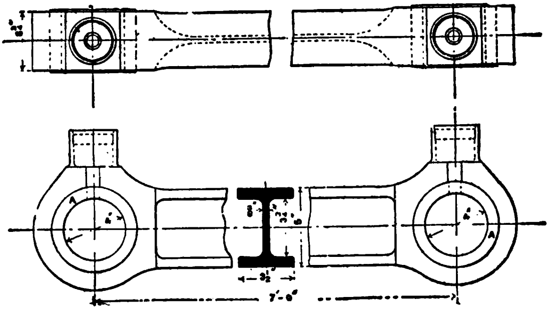


FIG 1,938 and 1,939.—Non-adjustable coupling rod. The bearings consist of bushings which are pressed into position and sometimes held from turning by a set screw. These brasses when worn are replaced by new ones bored to fit the pins.

The Valve Gear.—Formerly the so called Stephenson link motion was universally used on locomotives, but is being displaced by the Walschaerts valve gear. The reason for this is because the size and arrangements of parts in a modern locomotive make it difficult for the engineer to properly examine the eccentrics and link when the engine is on the road, and breakdowns are more frequent on this account. The conditions of service also tend to make it more and more difficult for engineers to give the close inspection and care which is demanded in other branches of engineering service with high speed machines.

The familiar link motion has been explained at such length in Chapter 10 on radial valve gears that hardly any further mention is necessary, except to show the general arrangement of parts, as in fig. 1,940, adapting the gear to locomotives.

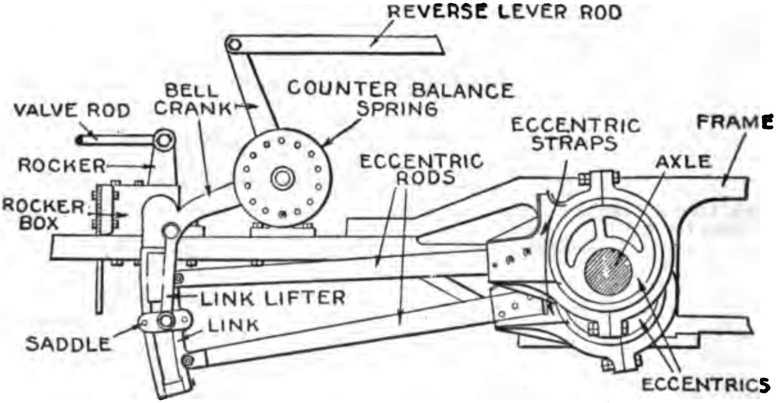


Fig. 1,940.—The so called Stephenson link motion. View showing general arrangement and design of parts and detail of frame, with names of parts.

The Walschaerts valve gear requires only one eccentric, or its equivalent, for each cylinder, to insure the movement of the valve, and the proper distribution of the steam for both forward and backward motion. While it is not possible to adjust the valve as readily with the Walschaerts gear as with the shifting link, for the reason that the parts and connections are not as susceptible to change, it is not as liable to become disarranged, and if correctly designed and fitted will give accurate results.

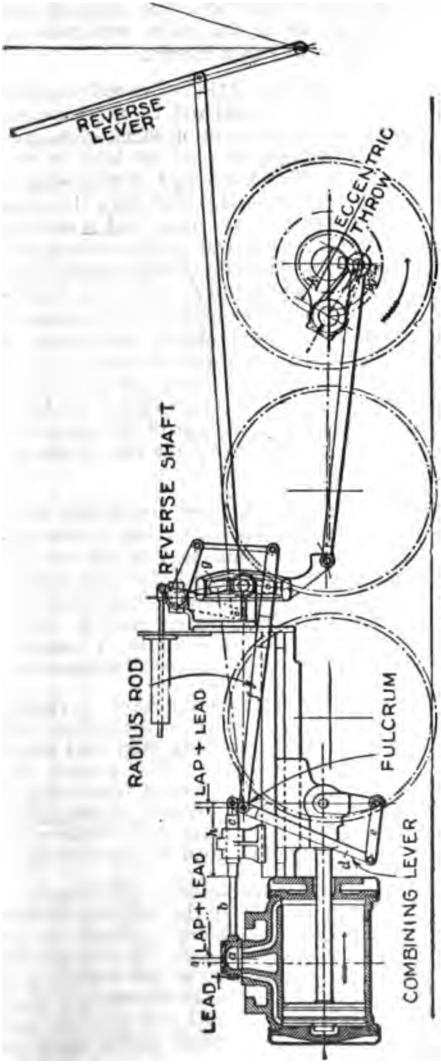
The eccentric is secured to the driving axle either directly or by a return crank from one of the crank pins.

The position of the eccentric or crank is such as to give the proper valve travel, the throw corresponding with the movement of the valve irrespective of its lap and lead, the angular advance of the eccentric being 0°.

The link is of any convenient form and is usually pivoted to a support on the engine frame or suspended from the guide bearer. The trunnion is rigid and there is no chance for twisting strains.

The link is actuated by the eccentric rod which is commonly attached to its lower extremity.

The sliding block in the link is secured to one end of the radius rod. The raising or lowering of this rod by means of the reversing shaft shifts the block from one end of the link to the other above or below



FIGS. 1,941.—Walschaerts valve gear with extended radius rod. Sectional view showing general arrangement and names of parts.

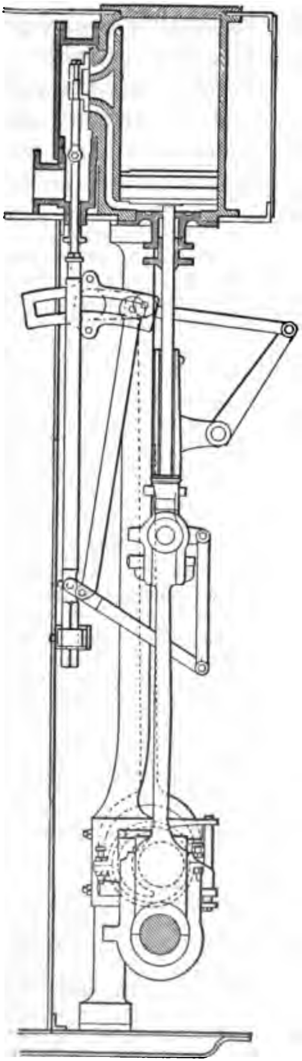


FIG. 1,942.—Walschaerts valve gear with return radius rod. In this arrangement the link is very near the cylinder.

pivoted connection; this reverses the movement of the valve with relation to that of the eccentric.

The end of the radius rod opposite the link is attached to a combining lever, the function of which is to give the required amount of lead to the valve. The lower end of this lever is connected to and travels with the cross head, while to the upper end is secured both the valve rod and the radius rod, one being placed above the other. The point at which the radius rod is attached to the combining lever becomes a fulcrum. The relative movement of the two ends of the lever must be such, that the full stroke of the cross head imparted to the lower end of the lever will give a movement of the upper end equivalent to twice the sum of the required lap and lead.

Under ordinary conditions with steam chest valves having outside admission, the connection or fulcrum between the radius rod and the combining lever is placed below the valve rod connection. With valves having inside admission, this fulcrum is usually placed above the valve rod connection.

The link should have a radius equal to the length of the radius rod. If this be so it will be seen that when the engine is on the dead center the link block can be moved from end to end of the link without altering the position of the valve with relation to the ports and the lead will be constant.

As any variation in the length or relative position of the link, the radius rod, and the combining lever or its connections, will necessarily change the resulting movement of the valve; it is essential, first, that the motion shall be correctly designed and plotted, and second, that the detail parts shall be

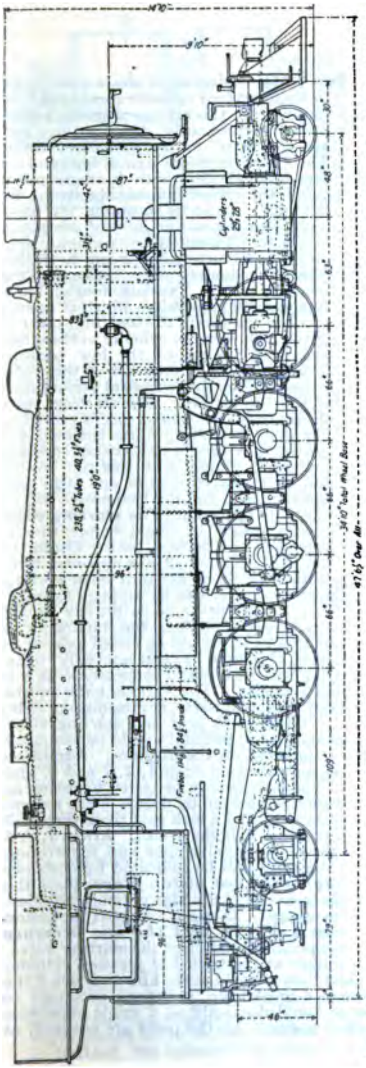


FIG. 1,943.—Chesapeake & Ohio ***Mikado*** locomotive. Maximum tractive power, 60,800 lbs.; weight on driving wheels, 243,000 lbs.; weight on leading truck, 27,000 lbs.; on trailing truck, 45,000 lbs.; total weight of engine, 315,000 lbs.; wheel base rigid, 16′ 6″; total wheel base of engine, 34′ 10″; cylinders 29×28; boiler pressure, 170 lbs.; boiler diam., front, 83¾, back, 96; fire box, 84¼×114⅛; tubes, 238, 2¼″ and 40, 5½″; length 19′; combustion chamber length 19½″; heating surface total, 4,051 sq. ft.; superheating surface, 832 sq. ft.; grate area, 66.7 sq. ft.; fuel, soft coal; water capacity, 9,000 gals.; coal, 15 tons; wheel base engine and tender, 67′ 10½″; height, 14′ 10″; width, 10′ 5″; length all over, 77′ 3⅞″.

NOTE.—***General instructions for Walschaerts valve gear*** (Baldwin). In setting this valve gear it must be remembered that two distinct motions are in combination: 1, the motion due to the cross head travel, and 2, the motion due to the eccentric throw. The cross head motion controls the lead, by moving the valve sufficiently to overcome its lap, plus the amount of lead in both front and back positions. The eccentric throw controls the travel and reversing operations. It will be seen that the movement due to the eccentric, without the cross head motion, would place the valve centrally over the ports when the piston is at the extreme end of the stroke. The combined effect of these two motions, when the parts are properly designed, gives the required movement of the valve, similar to that obtained by the use of a stationary link. ***To reverse*** the engine, the link block is moved from end to end of the link, instead of moving the link on the block. This operation is accomplished by means of a reversing shaft connected with a reversing lever in the cab. All parts of the Walschaerts gear should be correctly laid out and constructed from a diagram, as the proportions cannot be altered without seriously affecting the correct working of the device. The only part capable of variation in length is the eccentric rod, which connects the eccentric with the link. The rod may be slightly lengthened or shortened, to correct errors in location of the link center, from center of driving axle which carries the eccentric. When the engine is assembled, the throw of the eccentric should be checked up by the specifications, and any error should be at once reported in order that the mistake may be rectified by either correcting the position of the eccentric, or by a change in the design of the other parts to compensate for the error. In case of accident, if any of the rods or connections be broken it is advisable if possible to disconnect the eccentric rod. The combining lever should be uncoupled from the cross head and securely fastened in forward position.

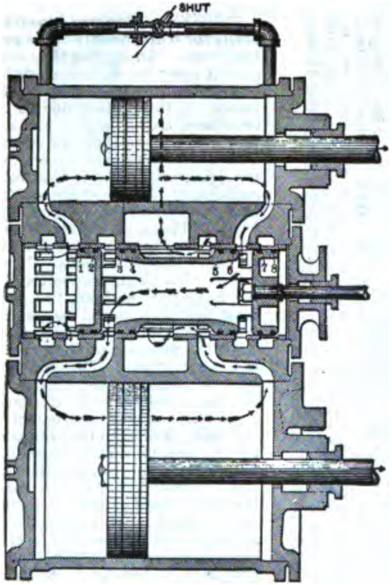
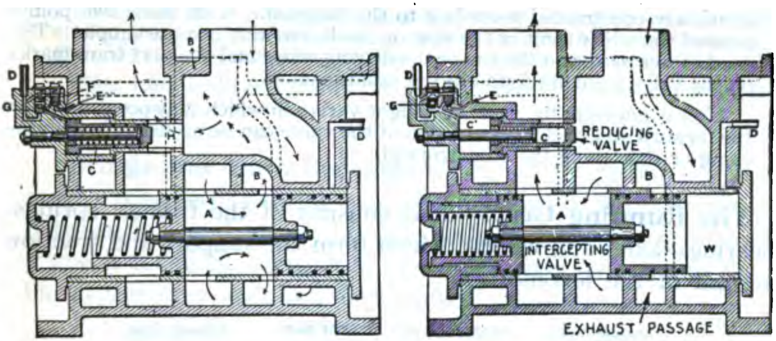


FIG. 1,944.—Diagram of steam distribution in Vauclain four cylinder compound locomotive. In this arrangement the pistons work synchronously, that is their rods are connected to the same cross head. The figure shows one pair of high and low pressure cylinders with a single piston valve, the arrows indicate the steam distribution. In order to obtain the maximum starting power in any compound locomotive it is necessary to employ some means of admitting live steam to the low pressure cylinder. The device for this purpose in the Vauclain four cylinder type is a by pass valve, which is opened to allow the steam to pass from one end of the high pressure cylinder to the other, and from thence to the low pressure cylinder. The proper use of this valve is essential to the successful performance of the locomotive. The starting valve should never be open at speeds exceeding five miles per hour, unless the engine is drifting down grade with a closed throttle. Failure to comply with this rule will result in excessive repairs and high fuel consumption, and the locomotive will be "logy" in operation. ***The starting valve*** should always be closed before the reverse lever is hooked back. Owing to the mild exhaust of a compound locomotive the fire is not torn when working the engine full stroke; hence in ascending grades, the reverse lever can be dropped forward, thus keeping up the speed without injury to the fire. If the engine be so heavily loaded that the speed drops to five miles per hour, and there is danger of stalling, the starting valve may be opened in order to keep the train moving; but this valve should be closed as soon as the ordinary difficulty is overcome. If a locomotive of this type exhaust unevenly, or be lame, a careful inspection should be made of the starting valve levers and connections, as it is important that the two valves open and close simultaneously. The various parts of the valve gear should also be carefully examined, as bent eccentric rods or transmission rods, loose rocker boxes, etc., invariably cause trouble. If the valve gear and starting valve rigging be in good condition, the valve packing rings and piston packing should be examined for leaks and blows. The valve packing rings are numbered on the diagram. Admission and release of steam to the high pressure cylinder is controlled by rings 1, 2, 7 and 8, and to the low pressure cylinder by rings 3, 4, 5 and 6. These rings may be tested for blows as follows: ***Rings 1, 2, 7 and 8.*** Place the valve in its middle position by means of the reverse lever, so that all the ports are uncovered. Open the throttle, and a leak past these rings will be shown by a steady escape of steam at the high pressure cylinder cocks. ***Rings 3, 4, 5 and 6.*** Place the reverse lever in full gear, with the starting valve open and the driving brakes applied. Open the throttle, and a leak in the rings will be indicated by a steady blow through the exhaust nozzle. ***To test the high pressure piston rings,*** place the engine at about quarter stroke, admitting steam to the front end of the high pressure cylinder. Keep the starting valve closed and the driving brakes applied. If steam leak past the piston it will escape in a steady stream at the front cylinder cock. ***To test the low pressure piston rings,*** keep the engine and valve in the same position, and open the starting valve. A leak past the rings will be indicated by a steady blow at the back low pressure cylinder cock. The testing of valves and pistons for leaks and blows should always be done when the cylinders are hot and well lubricated. If, because of a breakdown, it become necessary to disconnect a locomotive of this type, the engine is handled exactly as a single expansion locomotive. When the valve is placed in its central position, all the ports are covered, as in the case of an ordinary slide valve.

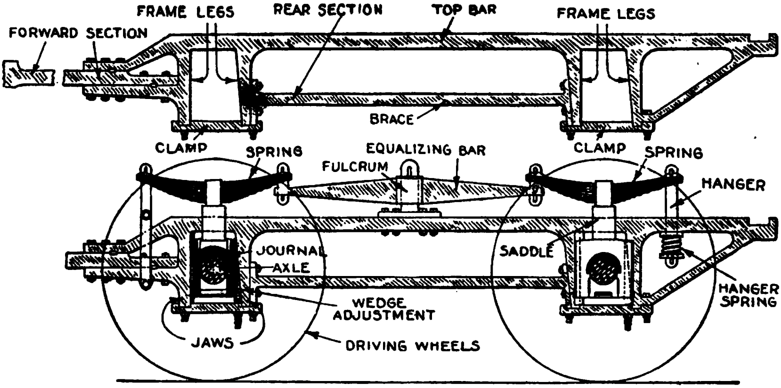


FIGS. 1,945 and 1,946.—Baldwin two cylinder compound valves, fig. 1,945, position when working single expansion; fig. 1,946, position when working compound. In the figures A is an "intercepting" valve; its movement in one direction is controlled by a spring, and in the other, by steam pressure. ***The function of the intercepting valve*** is to cause the exhaust steam from the high-pressure cylinder to be diverted, at the option of the engineer, either to the open air when working single expansion, or to the receiver when working compound. C, is a ***reducing valve*** also moved in one direction by a spiral spring and in the other by steam pressure. The function of this valve is, in its normal position, to admit live steam into the receiver at reduced pressure while the locomotive is working single expansion. When the locomotive is working compound, this valve automatically closes. A further function of the reducing valve is to regulate the pressure in the receiver so that the total pressure on the pistons of the high and low pressure cylinders may be equalized. The steam for controlling the operation of both intercepting and reducing valves is supplied through the pipes D, from the operating valve in the cab. When not permanently closed by pressure in the pipes D, the reducing valve C, is operated automatically by the pressure in the receiver. To this end the port E, is provided, communicating with the receiver and the space in front of the reducing valve; as the pressure rises the steam acts on the large end of the reducing valve, causing it to move backward and close the passage H, through which steam enters the receiver, and thus prevents an excess pressure of steam in the low pressure cylinder. Poppet valves F and G, are placed in connection with the port E, one to prevent the escape of steam from the receiver to the pipe D, when the locomotive is working single expansion, and the other to close the passage from pipe D, to the receiver when working compound. Normally the lever of the operating valve in the cab is in the position marked "simple." In this position no steam is allowed to enter the pipes D, and no pressure will be exerted on the intercepting and reducing valves in opposition to the springs, and they will assume the positions shown in fig. 1,945. The ports of the intercepting valve A, stand open to receive the exhaust steam from the high pressure cylinder and deliver it through the exhaust passage B, to the atmosphere. The reducing valve is open, admitting live steam through passage H, to the receiver and from there to the low pressure cylinder. ***The receiver pressure*** is governed by the automatic action of the reducing valve as previously explained. In this way the locomotive can be used single expansion in making up and starting trains for switching and slow running. At the will of the engineer the operating valve in the cab is moved to the position marked "compound." This admits steam to the pipes D, and through them to the valve chambers W and C′, changing the intercepting and reducing valves instantly and noiselessly to the positions shown in fig. 1,946. The exhaust from the high pressure cylinder is diverted to the receiver, the admission of live steam to the receiver is stopped by the closing of the passage H, and the locomotive is in position to work compound. A locomotive of the two cylinder compound type is tested for leaks or blows in the same manner as a single expansion engine. The tests should be made when the engine is working single expansion at slow speed, with the cylinders warm and well lubricated. ***In case of a break down,*** the engine can be disconnected as readily as a single expansion locomotive and in exactly the same manner; the main rod should be taken down, the cross head blocked and the valve placed in its central position to cover all ports. In all cases, regardless of which side is disabled, the intercepting valve must be in position for working single expansion.

accurately constructed according to the diagram. With these two points assured the adjustment of the gear on the locomotive is quite simple. The dead center marks on the rim of the driving wheel and the port tram marks on the valve stem are found in the usual manner.

After connecting the gear, any slight variation which may occur between the forward and backward position of the valve can be adjusted by lengthening or shortening the eccentric rod.

The Running Gear.—This consists of the frames, springs, bearings, axles and wheels, which form the support and traction system of the locomotive.



Figs. 1,947 and 1,948.—Detail of frames, springs, equalizer, and driving boxes, with names of parts.

Frames.—These are made of wrought iron bars three or four inches thick, and about the same in width. There are two frames, one on each side of the boiler.

Each frame is made in two sections, a forward member consisting of a single bar, and a rear member containing the guides for the bearings. The forward section is attached to the cylinder casting, which in turn is bolted to the smoke box of the boiler; there are diagonal braces, the lower ends of which are bolted to the bumper timber and to the frame, and the upper ends of which are bolted to the smoke box; there are also braces between the boiler and the frames.

The rear frame sections pass through expansion clamps bolted to the side of the fire box to prevent expansion and contraction of the boiler.

In addition, there are usually diagonal braces bolted above to the back end of the outer fire box sheet, at about the height of the crown sheet, their lower ends being bolted to the frames at the back ends.

Figs. 1,947 and 1,948 shows the general construction of the frames and names of parts.

Springs and Main Bearings.—The spring system of a locomotive consists of a spring over each driving box with connections between box and frame. Connection between the springs and frame is made by hangers at the outer ends and by equalizing bar at the inner ends as shown in fig. 1,948.

The driving or main springs are made of steel plates or leaves about ⅜ to 7/16 inch thick by 4 to 5 inches wide and of lengths from center to center of hangers, varying from 36 to 48 inches.

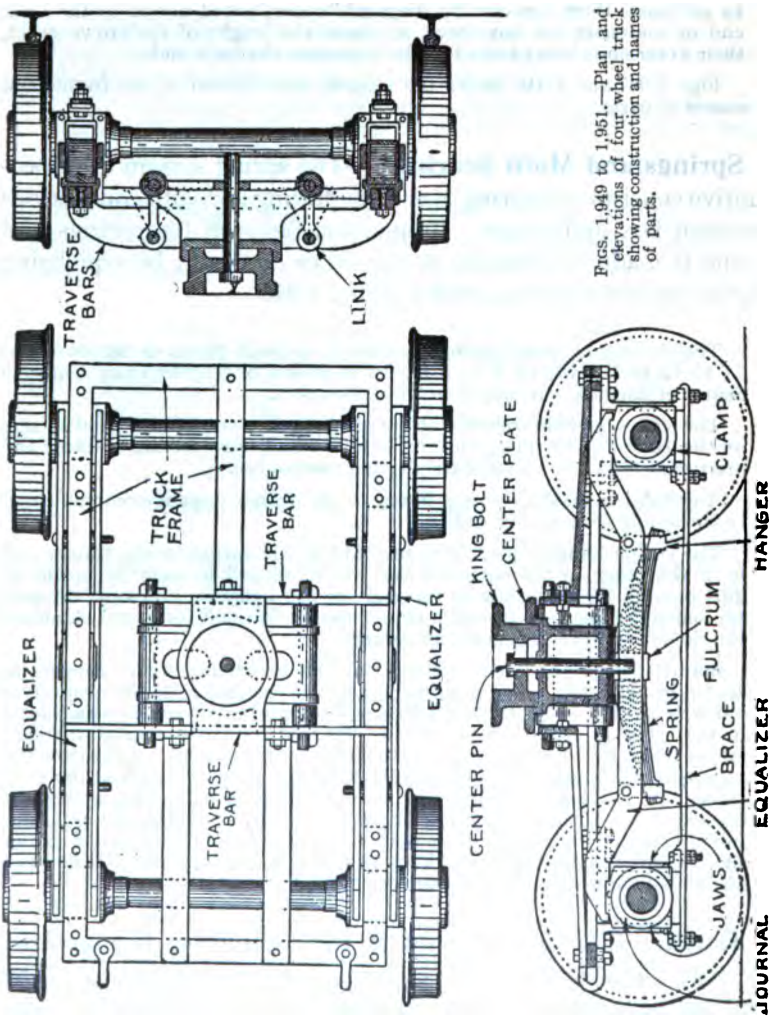
The leaves are held together at the center by a wrought iron band shrunk on the leaves. The springs rest on saddles which span the top of frame and are supported on the tops of the driving journal boxes.

Any motion of the driving boxes in the frame jaws is communicated to the spring through the saddle.

The spring hangers pass from one end of the spring to the frame, and from the other to the equalizer and are connected to each by means of gibs passing through slots in the ends of the hangers. In some designs, the hangers pass over the ends of the springs in hook form and eliminate the slots in both the hangers and springs.

Spiral springs are sometimes used at the lower end of the hangers as shown in fig. 1,948. The equalizer is a lever extending from the spring over one axle to that of another and having a fulcrum at its center which rests in the frame or on one of the boxes. Its function is to equalize the load on the boxes with which it connects and also to transmit the shocks from the rails to the springs. By shifting the fulcrum more or less load may be placed on the springs connected by the equalizer. By this system of equalizer and spring suspension on each side of the boiler, the equalizer fulcrums represent the two rear points of a tripod, while the center pin of the truck is the third point, making a three point bearing which makes the system adjustable to the most uneven track.

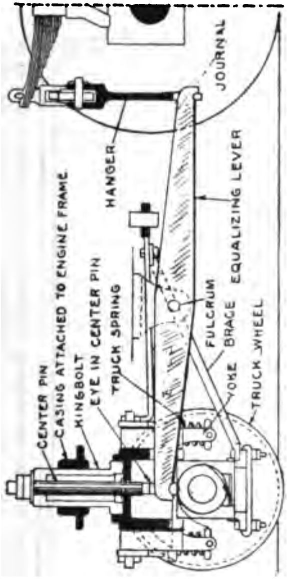
Trucks.—The forward end of the locomotive is supported by a truck which may have either two or four wheels. The four wheel truck is the more extensively used and has a plate spring similar to a driving spring between each pair of wheels.

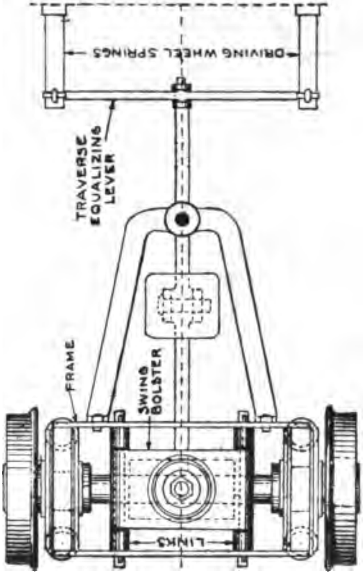


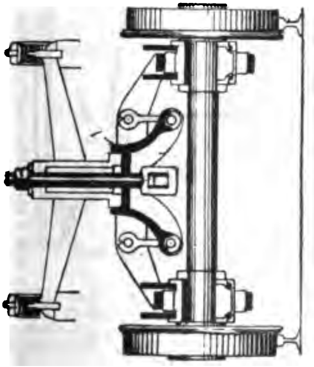
FIGS. 1,949 to 1,951.—Plan and elevations of four wheel truck showing construction and names of parts.

placed in an inverted position with the band resting on the underside of the truck frame and the ends resting in strong spring hangers, held between a pair of equalizers, the butt ends of which are carried on the tops of the truck boxes. The bottom leaf of the springs is formed into a rounding seat to receive the spring hangers.

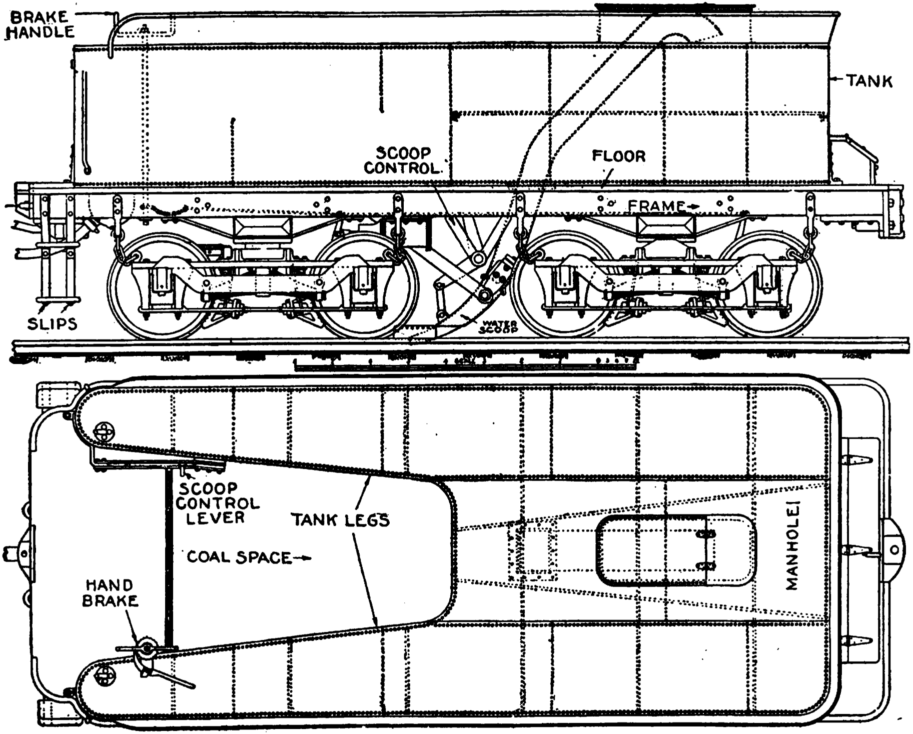
Figs. 1,949 to 1,954 show the details of both types of truck.







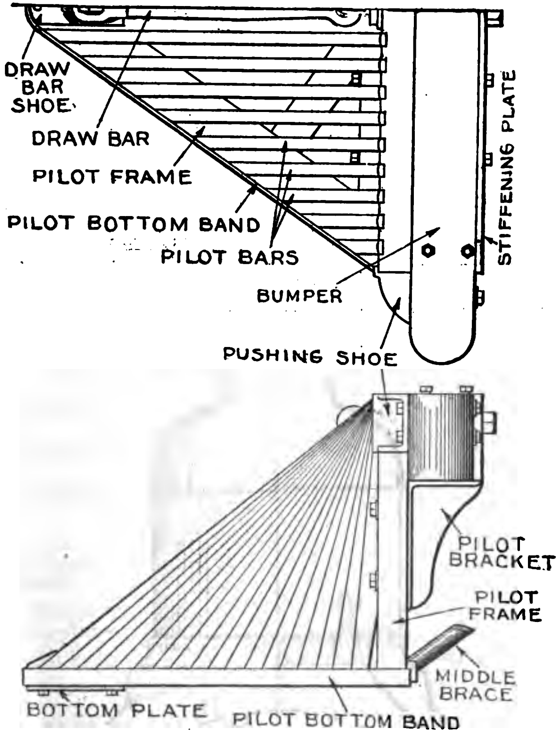
Figs. 1,952 to 1,954.—Plan and elevation of Bissell or Pony (two wheel) truck showing construction and names of parts.



FIGS. 1,955 and 1,956.—Plan and elevation of locomotive tender showing construction details and names of parts. Among the details shown is the water scoop (seen in elevation in fig. 1,955), for replenishing the supply without stopping.

Tender.—The tender is a vehicle separate and distinct from the locomotive proper. It carries the fuel and water and is coupled to the engine.

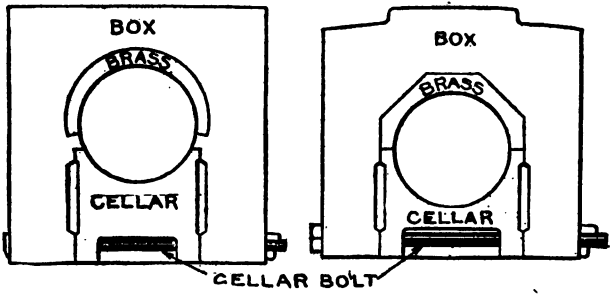
That part of the tender which contains the water, that is the tank, is of U shape in plan, formed of two legs which are at the sides, and a larger rectangular space at the rear, having a capacity ordinarily of from four to nine thousand gallons of water.



Figs. 1,957 and 1,958.–Pilot and bumper. The pilot consists of a frame having a V shaped base and a V shaped back, attached to the bumper timber and tending to throw to one side of the track any comparatively light object which may strike thereon. The pilot is popularly known as the "cow catcher."

The coal is contained in the space between the two legs which is called the coal pit. The coal capacity is from six tons for the smaller tenders to sixteen tons for the larger ones.

The tender is mounted on two four wheel trucks. On some tenders provision is made for filling the water tank from a track tank. The device consists of a spout of irregular curved shape, the upper end of which is located in the manhole and curved downward, while the lower end is fitted with a rectangular mouth which is pivoted so as to lift clear of the drop to the track tank. The moveable end of the spout is opened by compressed air.



Figs. 1,959 and 1,960.—Driving and truck journal boxes.

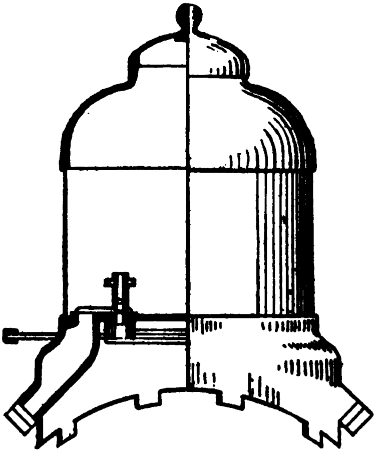


Fig. 1,961.—Detail of sand box. It is placed on top of the boiler usually between the steam dome and smoke stack.

The Air Brake.—In this type of brake, which operates by the expansive property of compressed air, the air is compressed by a suitable pump located upon the engine and is stored until needed for use. When it is necessary to apply the brake, a portion of the stored air is allowed to pass into the brake cylinder. This cylinder is fitted with a piston which the escaping air moves outward. It is so connected with the brakes that its movement is communicated to the shoes and they are applied.

The kind of air brake now used is known as the automatic, the term meaning that in case of accident such as the parting of a train, bursting of an air pipe, etc., the brake is automatically applied. The elementary diagram fig. 1,962 illustrates the essential parts of the apparatus and principles of operation.

The two makes of air brake chiefly used are the Westinghouse and the New York. These operate on the same principle and differ only in the mechanical details.

The automatic air brake, as already mentioned is one so constructed that the brake will be applied *automatically* in case of accident. To accomplish this an auxiliary reservoir and a distributing or triple valve is necessary in each car.

The functions of the triple valve are briefly

1. When charging and maintaining the pressure in the brake system:

   a. To permit air to flow from the brake pipe to the auxiliary reservoir;

   b. To prevent air flowing from the auxiliary reservoir to the brake cylinder;

   c. To keep the brake cylinder open to the atmosphere.

2. When applying the brakes:

   a. To close communication from the brake pipe to the auxiliary reservoir;

b. To close communication from the brake cylinder to the atmosphere.
c. To permit air to flow from the auxiliary reservoir to the brake cylinder.

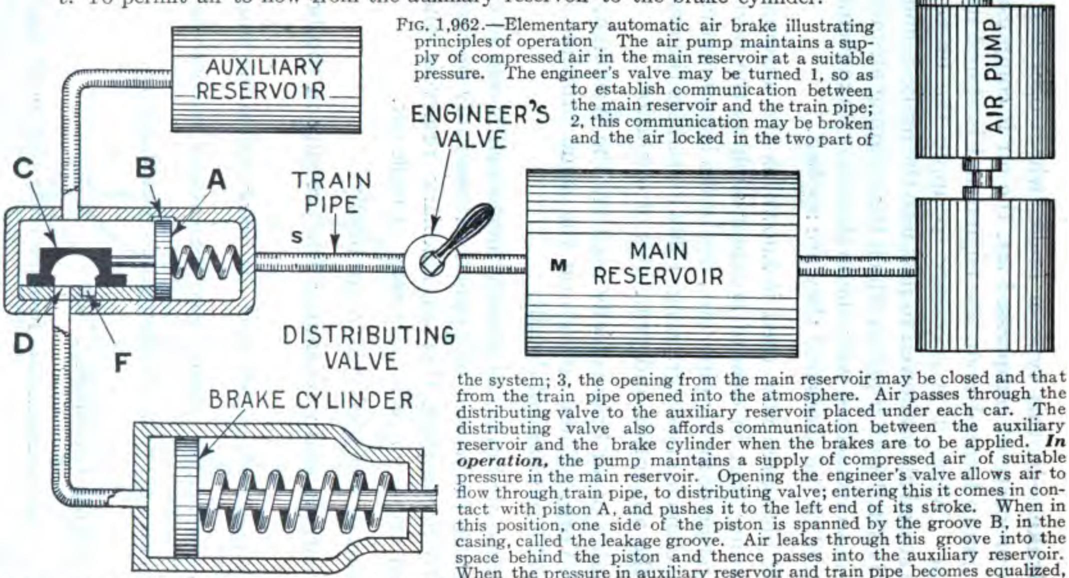


FIG. 1,962.—Elementary automatic air brake illustrating principles of operation The air pump maintains a supply of compressed air in the main reservoir at a suitable pressure. The engineer's valve may be turned 1, so as to establish communication between the main reservoir and the train pipe; 2, this communication may be broken and the air locked in the two part of the system; 3, the opening from the main reservoir may be closed and that from the train pipe opened into the atmosphere. Air passes through the distributing valve to the auxiliary reservoir placed under each car. The distributing valve also affords communication between the auxiliary reservoir and the brake cylinder when the brakes are to be applied. ***In operation,*** the pump maintains a supply of compressed air of suitable pressure in the main reservoir. Opening the engineer's valve allows air to flow through train pipe, to distributing valve; entering this it comes in contact with piston A, and pushes it to the left end of its stroke. When in this position, one side of the piston is spanned by the groove B, in the casing, called the leakage groove. Air leaks through this groove into the space behind the piston and thence passes into the auxiliary reservoir. When the pressure in auxiliary reservoir and train pipe becomes equalized, the apparatus is said to be charged and the brakes can be applied by turning the engineer's valve so that the flow from the main reservoir is stopped and the air in the train pipe is exhausted into the atmosphere; this causes excess pressure on left face of piston A. The air then starts to rush out from auxiliary reservoir through leakage groove, but as it cannot escape rapidly enough to supply the loss in D, the piston A, is moved to the right moving with it the attached slide valve C, which uncovers the port D, leading to the brake cylinder. This allows air to flow into the brake cylinder and apply the brake. Meanwhile the piston A, has traveled to the right, and beyond the end of the leakage groove B, stopping the flow of air from the auxiliary reservoir to the train pipe, thus setting the brakes, they will remain set until released. To release brakes the engineer re-establishes communication between the main reservoir M, and the train pipe S. Air again enters the triple valve. The piston A, is pushed to the left, thus C, is shifted so air flowing from auxiliary reservoir into the brake cylinder is stopped and the brake cylinder vented to atmosphere by means of the passage beneath valve which connects port D, and exhaust port F, the latter being open to the atmosphere.

3. When holding the brakes applied:

a. To close all communication between the brake pipe, auxiliary reservoir, brake cylinder and atmosphere.

4. When releasing the brakes and recharging the system:

a. To open communication from the brake cylinder to the atmosphere;
b. To permit air to flow from the brake pipe to the auxiliary reservoir;
c. To prevent air flowing from auxiliary reservoir to the brake cylinder.

The Westinghouse quick action automatic air brake is the same as the plain automatic but with the additional feature that the triple valve is so modified that when a relatively quick reduction in brake pipe pressure is made, it also opens a direct communication from the brake pipe through the triple valve to the brake cylinder. This not only increases the brake cylinder pressure in proportion to the amount of air flowing into it from the brake pipe, but by venting air from the brake pipe locally on each car, hastens and increases the effect of the reduction made at the brake valve; the net result being to greatly shorten the time from the movement of the brake valve handle until a full brake application is obtained on the entire train, and to increase the total braking power obtainable by such operation (called an emergency application of the brakes), about 20 per cent over the maximum obtainable during ordinary operations (called service applications of the brake), or when using the plain automatic brake. This difference is due solely to the construction of the triple valves, which are called respectively the plain and quick action triple valve.

For all ordinary (service) applications of the brake, the operation of the two triple valves is identical. The foregoing relates to apparatus which for years has been and is now being used on many railroads. It should be understood, however, that much of the apparatus has been superseded by later forms of the same device or entirely new devices or combinations; for example, the entire engine and tender equipment is superseded by the "ET" locomotive brake equipment.

The Westinghouse type "ET" locomotive brake equipment consists of the following parts:

1. The air compressor;
To compress the air.

2. The main reservoir;
To store and cool the air and to collect dirt and water.

3. A duplex compressor governor;

To control the compressor when the pressures for which it is regulated are obtained.

4. A distributing valve, and small double chamber reservoir to which it is attached;

Located on the locomotive, and performs the function of triple valves, auxiliary reservoirs, double check valves, high speed reducing valves, etc.

5. Two brake valves;

The *automatic* to operate the locomotive and train brakes, and the *independent* to operate the locomotive brakes only.

6. A feed valve;

To regulate the brake pipe pressure.

7. A reducing valve;

To reduce the pressure for the independent brake valve and for the air signal system when used.

8. Two duplex air gauges;

One to indicate equalizing reservoir and main reservoir pressures; the other, to indicate brake pipe and locomotive brake cylinder pressures.

9. Driver, tender, and truck brake cylinders, cut out cocks, centrifugal dirt collectors, hose couplings, fittings, etc., incidental to the piping for purposes readily understood.

The following are the names of the various pipes:

1. Discharge pipe;

Connects the air compressor to the first main reservoir.

2. Connecting pipe;

Connects the two main reservoirs.

3. Main reservoir pipe;

Connects the second main reservoir to the automatic brake valve, distributing valve, feed valve, reducing valve, and compressor governor.

4. Feed valve pipe;

Connects the feed valve to the automatic brake valve.

5. *Excess pressure pipe;*

Connects the feed valve pipe to the upper connection of the excess pressure head of the compressor governor.

6. *Excess pressure operating pipe;*

Connects the automatic brake valve to the lower connection of the excess pressure head of the compressor governor.

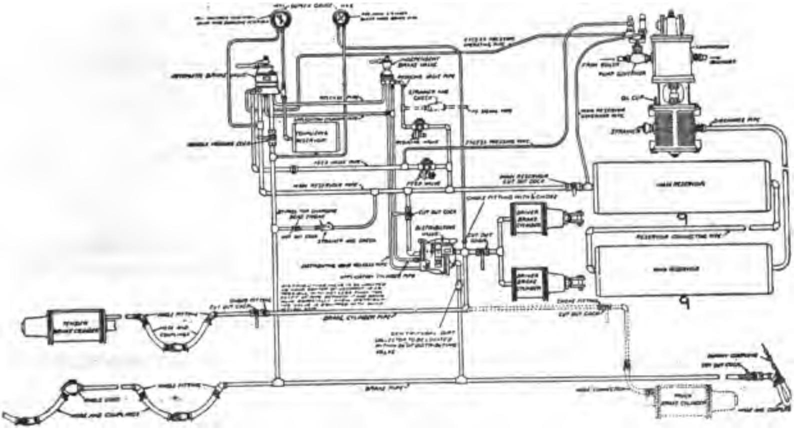


FIG. 1,963.—Diagram showing assembly of parts and pipes of Westinghouse "ET" locomotive air brake equipment with names of parts and pipes. *The compressor* passes as usual to the main reservoirs and the main reservoir pipe. The main reservoir cut out cock is to cut off and vent the air from the main reservoir pipe, when removing any of the apparatus except the governor. The end toward the main reservoir is tapped for a connection to the *compressor governor*. Before this cock is closed the double heading cock should be closed, and the brake valve handle placed in release position. This is to prevent the slide valve of the feed valve, and the rotary valve of the brake valve, being lifted from their seats. Beyond the main reservoir cut out cock, the main reservoir pipe has four branches, one of which runs to the automatic brake valve, one to the feed valve, one to the reducing valve, and one to the distributing valve. As a result, the automatic brake valve receives air from the main reservoirs in two ways, one direct and the other through the feed valve. The feed valve pipe from the feed valve to the automatic brake valve has a branch to the top of the excess pressure head of the duplex compressor governor. The third branch of the main reservoir pipe connects with the reducing valve. Air at the pressure for which this valve is set (45 pounds) is supplied to the independent brake valve through the reducing valve pipe When the air signal system is installed, it is connected to the reducing valve pipe, in which case the reducing valve also takes the place of the signal reducing valve formerly employed. In the branch pipe supplying the air signal system is placed a combined strainer, check valve, and choke fitting. The strainer prevents any dirt reaching the check valve and choke fitting. The check valve prevents air flowing back from the signal pipe when the independent brake is applied. The choke fitting prevents the reducing valve raising the signal pipe pressure so quickly as to destroy the operation of the signal. *The distributing valve* has five pipe connections, made through the end of the double chamber reservoir, three on the left, and two on the right. Of the three on the left, the upper is the supply from the main reservoir; the intermediate is the application cylinder pipe, leading to the independent and automatic brake valves; and the lower is the distributing valve release pipe, leading through the independent brake valve, when the handle is in *running* position, to the automatic

7. ***Reducing valve pipe;***
   Connects the reducing valve to the independent brake valve, and to the signal system when used.

8. ***Brake pipe;***
   Connects the automatic brake valve with the distributing valve and all triple valves on the cars in the train.

9. ***Brake cylinder pipe;***
   Connects the distributing valve with the driver, tender and truck brake cylinders.

10. ***Application cylinder pipe;***
    Connects the application cylinder of the distributing valve to the independent and automatic brake valves.

11. ***Distributing valve release pipe;***
    Connects the application cylinder exhaust port of the distributing valve to the automatic brake valve through the independent brake valve.

The diagram fig. 1,963 shows the assembly and arrangement of the parts and piping just described.

Principles of Operation.—It should be noted that the principles governing operation of the type "ET" equipment are the same as those on previous equipments; the difference consists in the means for supplying the air pressure to the brake cylinders.

FIG. 1,963—*Text Continued.*

brake valve. Of the two on the right, the lower is the brake pipe branch connection, and the upper is the brake cylinder pipe, branching to all brake cylinders on the engine and tender. In this pipe are placed cocks for cutting out the brake cylinders when necessary, and in the engine truck and tender brake cylinder cut out cocks are placed choke fittings to prevent serious loss of main reservoir air and the release of the other locomotive brakes during a stop, in case of burst brake cylinder hose. The two duplex air gauges are connected as follows: Gauge No. 1; red hand, to main reservoir pipe under the automatic brake valve; black hand, to equalizing reservoir pipe tee of the automatic brake valve. Gauge No. 2; red hand, to the brake cylinder pipe; black hand, to the brake pipe below the double heading cock. The amount of reduction made during an automatic application is indicated by the black hand of gauge No. 1, the black hand of gauge No. 2 is to show the brake pipe pressure when the engine is second in double heading, or when a helper. The automatic brake valve connections, other than already mentioned, are the brake pipe, the main reservoir, the equalizing reservoir, and the lower connection to the excess pressure head of the compressor governor.

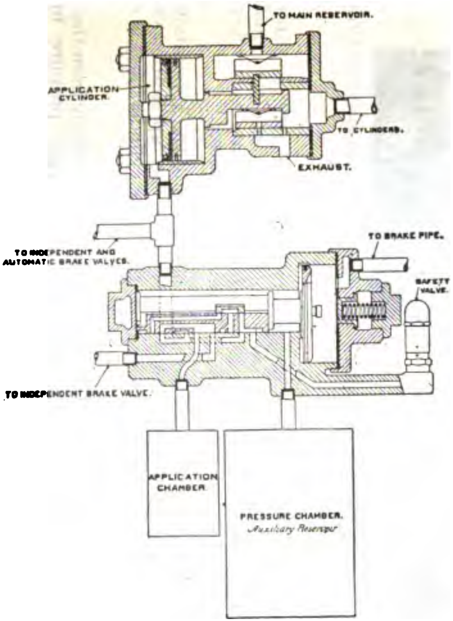
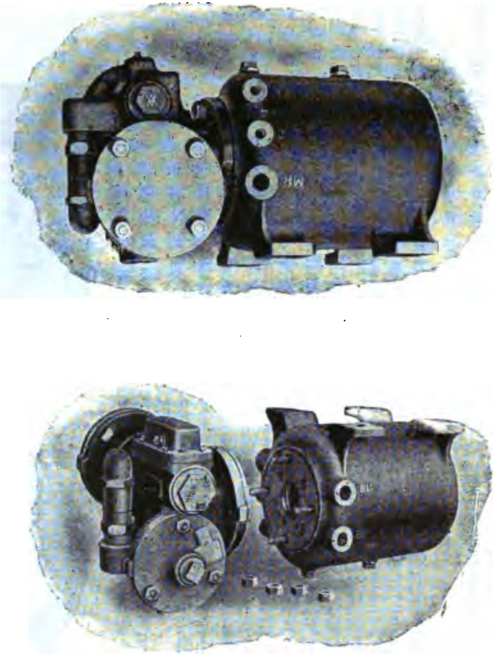


Fig 1,964.—Diagrammatic view showing the essential parts of Westinghouse "ET" distributing valve and double chamber reservoir.



Figs. 1,965 to 1,971—Westinghouse No. 6 distributing valve and double chamber reservoir *connections*: M R, main reservoir pipe; 4, distributing valve release pipe; 2, application cylinder pipe; 5, C Y L S, brake cylinder pipe; BP, brake pipe.

Instead of a triple valve and auxiliary reservoir for each of the engine and tender equipments, the distributing valve is made to supply all brake cylinders.

The distributing valve consists of two portions called the "equalizing portion" and "application portion." It is connected to a "double chamber reservoir," the two chambers of which are called respectively the "pressure chamber" and the "application chamber." The latter is ordinarily connected to the application portion of the distributing valve in such a way as to enlarge the volume of that part of it called the *application cylinder*, as shown in fig. 1,965. The connections between these parts as well as their operation, may be compared with that of a miniature brake set,—the equalizing portion representing the triple valve; the pressure chamber, the auxiliary reservoir; and the application portion always having practically the same pressure in its cylinders as that in the brake cylinders. This is shown by the diagrammatic illustration in fig. 1,964.

For convenience, compactness and security, they are combined in one device as shown, partly dissembled, in fig. 1,965.

Figs. 1,972 and 1,973.—Westinghouse duplex gauges; fig. 1,972 large air gauge; fig. 1,973 small air gauge.

The equalizing portion and pressure chamber are used in automatic applications only; reductions of brake pipe pressure cause the equalizing valve to connect the pressure chamber to the application chamber and cylinder, allowing air to flow from the former to the latter.

The upper slide valve connected to the piston rod of the application portion, admits air to the brake cylinders and is called the "application valve," while the lower one releases the air from the brake cylinders and is called the "exhaust valve."

Since the air admitted to the brake cylinders comes directly from the main reservoirs, the supply is practically unlimited. Any pressure in the application cylinder will force the application piston to close the exhaust valve, open the application valve, and admit air from the main reservoirs to the locomotive brake cylinder until their pressure equals or slightly exceeds that in the application cylinder; whereupon the application piston and valve will be returned to lap position, closing the application valve. Also any variation of application cylinder pressure will be exactly duplicated in the locomotive brake cylinders, and the resulting pressure maintained regardless of any brake cylinder leakage. The operation of this locomotive brake, therefore, depends upon the admitting of air to and the releasing of air from the application cylinder; in independent applications, directly by means of the independent brake valve in automatic applications, by means of the equalizing portion and the air pressure stored in the pressure chamber.

The well known principle embodied in the quick action triple valve, by which it gives a high braking power in emergency

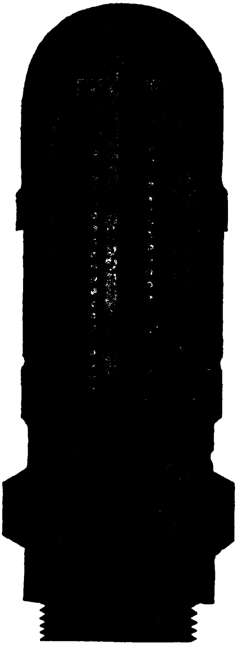
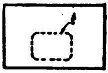
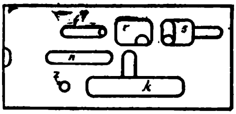


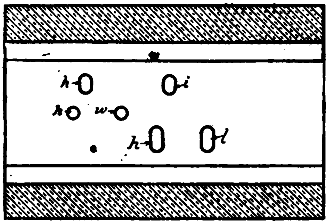
FIG. 1,974.—Westinghouse E-6 safety valve. It is unlike the ordinary safety valve, as its construction is such as to cause it to close quickly with a "pop" action, insuring its seating firmly. It is sensitive in operation and responds to slight differences of pressure. The names of the parts are: 2, body; 3, cap nut; 4, valve; 5, valve stem; 6, spring; 7, adjusting nut. Valve 4 is held to its seat by the compression of spring 6, between the flange of the stem and adjusting nut 7. When the air pressure below valve 4, is greater than the force exerted by the spring, it rises, and as a larger area is then exposed, its movement upward is very quick, being guided by the brass bush in the body 2. Two ports are drilled in this bush upward to the spring chamber; and two outward through the body to the atmosphere, although only one of each of these is shown in the cut. As the valve moves upward, its lift is determined by the stem 5 striking cap nut 3. It closes the two vertical ports in the bush connecting the valve and spring chambers, and opens the lower ports to the atmosphere. As the air pressure below valve 4, decreases, and the compression of the spring forces the stem and valve downward, the valve restricts the lower ports to the atmosphere and opens those between the valve and spring chambers. The discharge air pressure then has access to the spring chamber. This chamber is always connected to the atmosphere by two small holes through the body, 2; the air from the valve chamber enters more rapidly than it can escape through these holes, causing pressure to accumulate above the valve seat and assist the spring to close it with the "pop" action before mentioned. The safety valve is adjusted by removing cap nut 3, and screwing up or down the adjusting nut 7. After the proper adjustment is made, cap nut 3, must be replaced and securely tightened, and the valve operated a few times. Particular attention must be given to see that the holes in the valve body are always open, and that they are not changed in size, especially the two upper holes. This safety valve should be adjusted for 68 pounds. Like all adjustable devices, the safety valve is most easily and accurately adjusted when the work is done on a shop testing rack.

applications, and a sufficiently lower one in full service applications, to provide a desired protection against wheel sliding, is embodied in the distributing valve. This is accomplished by cutting off the application chamber from the application cylinder in all emergency applications. In such applications, the pressure chamber has to fill the small volume of the application cylinder only, thus giving a high equalization, and a correspondingly high brake cylinder pressure. In service applications, it must fill the same volume combined with that of the application chamber, thus giving a lower equalization and correspondingly lower brake cylinder pressure.

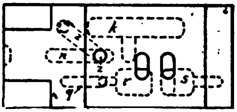

PLAN OF GRADUATING VALVE.



FACE OF SLIDE VALVE.



PLAN OF SLIDE VALVE SEAT.



PLAN OF SLIDE VALVE.

FIGS. 1,975 to 1,978—Details of Westinghouse No. 6 distributing valve. Part **h**, leads to the application cylinder, automatic brake valve, and independent brake valve; **w**, to the application chamber; **i**, to the distributing valve release pipe; **l**, to the safety valve.

The Distributing Valve.—This valve is the important feature of the ET equipment. Figs. 1,965 to 1,971 show photographic views of the valve and its double chamber reservoir, the pipe connection being plainly shown, the safety valve shown in fig. 1,974 is an essential part of the distributing valve. Figs. 1,975 to 1,978 show detail of graduating valve, equalizing valve and equalizing valve seat of the distributing valve.

Automatic Operation of the Distributing Valve.—To simplify the tracing of the ports and connections, the various positions of this valve are illustrated in the accompanying illustration, that is, the valve is distorted to show the parts differently than actually constructed with the object of explaining the operation clearly instead of showing exactly how they are designed. The chambers of the reservoir are for convenience indicated at the bottom as a portion of the valve itself. In fig. 1,974, equalizing piston 26, graduating valve 28, and equalizing slide valve 31, are shown as actually constructed. But as there are ports in the valves which cannot thus be clearly indicated, the diagrammatic illustrations show each slide valve considerably elongated so as to make all the ports appear in one plane, with similar treatment of the equalizing valve seat. Figs. 1,975 to 1,978 show the correct location of these ports.

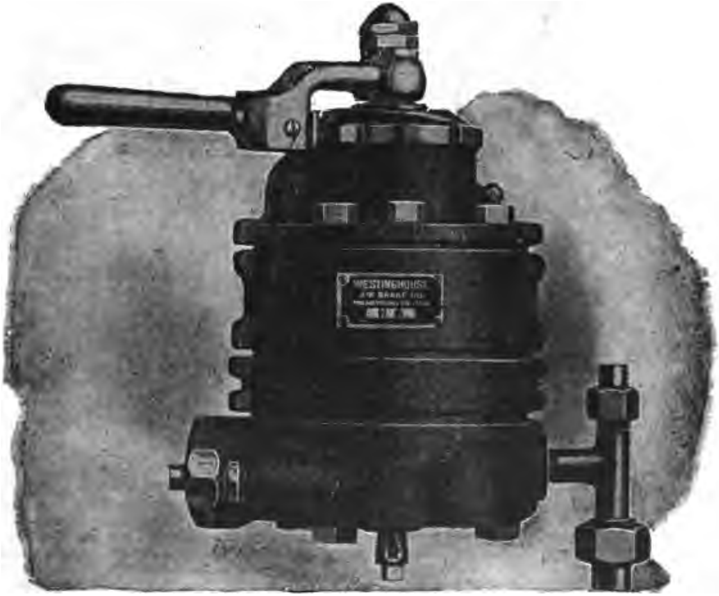


FIG. 1,979.—Westinghouse H-6 automatic brake valve.

Automatic Brake Valve. This brake valve, although modelled to a considerable extent upon the principles of previous valves, is necessarily different in detail, since it not only performs all the functions of such types but also those absolutely necessary to obtain all the desirable operating features of the No. 6 distributing valve.

Fig. 1,979 is taken from a photograph of this brake valve, while figs. 1,981 and 1,982 show two views, the upper one being a plan view with section through the rotary valve chamber, the rotary valve being removed; the lower one a vertical section. In these views the pipe connections are indicated.

Figs. 1,983 and 1,984 show two views of this valve similar to those of figs. 1,981 and 1,982, with the addition of a plan or top view of the rotary valve. The six positions of the brake valve handle are, beginning at the extreme left, release, running, holding, lap, service, and emergency.

In describing the operation of the brake valve, it will be more readily understood if the positions are taken up in the order in which they are most generally used, rather than their regular order as mentioned previously.

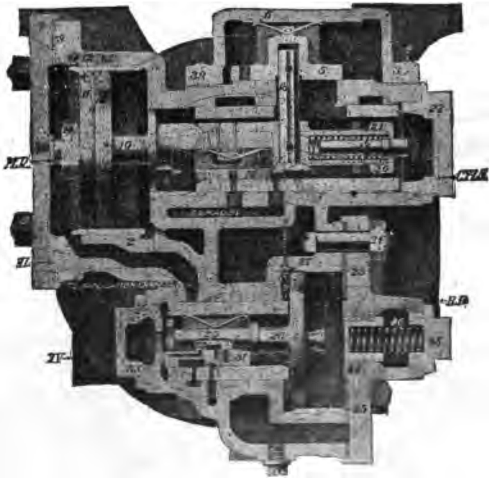


FIG. 1,980.—Westinghouse No. 6 distributing valve. *Connections*: M R, main reservoir pipe; IV, distributing valve release pipe; 11, application cylinder pipe; CYLS, brake cylinder pipe; B P, brake pipe. ***Parts;*** *2*, body; *3*, application valve cover; *4*, cover screw; *5*, application valve; *6*, application valve spring; *7*, application cylinder cover; *8*, cylinder cover bolt and nut; *9*, cylinder cover gasket; *10*, application piston; *11*, piston follower; *12*, packing leather expander; *13*, packing leather; *14*, application piston nut; *15*, application piston packing ring; *16*, exhaust valve; *17*, exhaust valve spring; *18*, application valve pin; *19*, application piston graduating stem; *20*, application piston graduating spring; *21*, graduating stem nut; *22*, upper cap nut; *23*, equalizing cylinder cap; *24*, cylinder cap bolt and nut; *25*, cylinder cap gasket; *26*, equalizing piston; *27*, equalizing piston ring; *28*, graduating valve; *29*, graduating valve spring; *31*, equalizing valve; *32*, equalizing valve spring; *33*, lower cap nut; *34*, safety valve; *35*, double chamber reservoir; *36*, reservoir stud and nut; *37*, reservoir drain plug; *39*, application valve cover gasket; *40*, application piston cotter; *41*, distributing valve gasket (located where distributing valve bolts to reservoir, not shown); *42*, oil plug; *43*, safety valve air strainer; *44*, equalizing piston graduating sleeve; *45*, equalizing piston graduating spring nut; *46*, equalizing piston graduating spring.

Charging and Release Position.—The purpose of this position is to provide a large and direct passage from the main reservoir to the brake pipe, to permit a rapid flow of air into the latter to: 1, charge the train brake system; 2, quickly release and recharge the brakes; but 3, not release locomotive brakes, if they be applied.

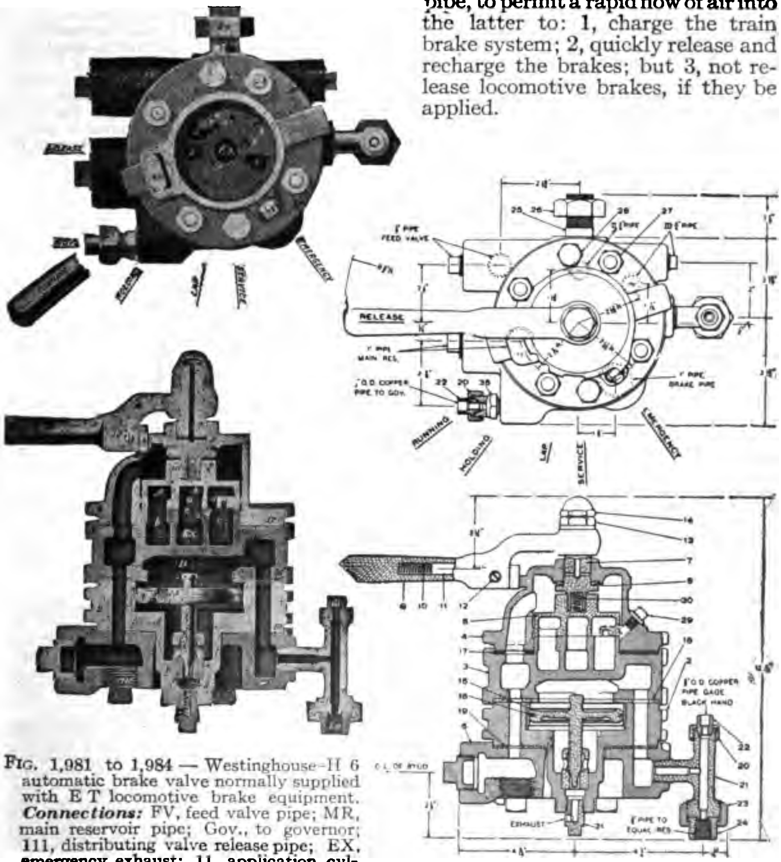


Fig. 1,981 to 1,984 — Westinghouse-H 6 automatic brake valve normally supplied with E T locomotive brake equipment. ***Connections:*** FV, feed valve pipe; MR, main reservoir pipe; Gov., to governor; 111, distributing valve release pipe; EX, emergency exhaust; 11, application cylinder pipe; BP, brake pipe; G A, duplex air gauge; E R, equalizing reservoir; B P E, service exhaust. ***Parts;*** 2, bottom case; 3, rotary valve seat; 4, top case; 5, pipe bracket; 6, rotary valve; 7, rotary valve key; 8, key washer; 9, handle; 10, handle latch spring; 11, handle latch; 12, handle latch screw; 13, handle nut; 14, handle lock nut; 15, equalizing piston; 16, equalizing piston packing ring; 17, valve seat upper gasket; 18, valve seat lower gasket; 19, pipe bracket gasket; 20, small union nut; 21, brake valve tee; 22, small union swivel; 23, large union nut; 24, large union swivel; 25, bracket stud; 26, bracket stud nut; 27, bolt and nut; 28, cap screw; 29, oil plug; 30, rotary valve spring; 31, service exhaust fitting; 35, governor union stud.

Air at the main reservoir pressure flows through port *a* in the rotary valve and port *b* in the valve seat to the brake pipe. At the same time, port *j* in the rotary valve registers with equalizing port *g* in the valve seat, permitting air at main reservoir pressure to enter chamber D above the equalizing piston.

If the handle were allowed to remain in this position, the brake system would be charged to main reservoir pressure. To avoid this, the handle must be moved to ***running*** or ***holding*** position. To prevent the engineer forgetting this, a small port discharges feed valve pipe air to the atmosphere in release position. Cavity *f* in the rotary valve connects port *d* with warning port *r* in the seat and allows a small quantity of air to escape into the exhaust cavity EX, which makes sufficient noise, to attract the engineer's attention to the position in which the valve handle is standing. The small groove in the face of the rotary valve which connects with port *s*, extends to port *p* in the valve seat, allowing main reservoir pressure to flow to the lower connection of the excess-pressure head of the compressor governor.

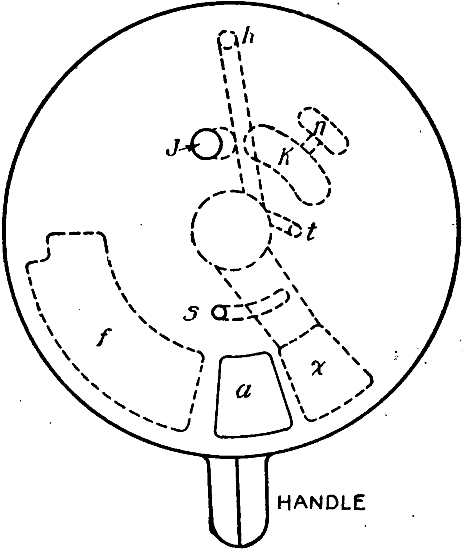


FIG. 1,985.—Detail of rotary valve of Westinghouse H-6 automatic brake valve. In the figure *a*, *j* and *s*, are ports extending directly through it, the latter connecting with a groove in the face; *f* and *k* are cavities in the valve face; *o* is the exhaust cavity; *x* and *t* are ports in the face of the valve connecting by cored passages with *o*; *h* is a port extending from the face over cavity *k* and connecting with exhaust cavity *o*; *n* is a groove in the face having a small port which connects through a cavity in the valve with cavity *k*. Referring to the ports in the rotary valve seats (fig. 1,981) *d* leads to the feed valve pipe; *b* and *c* lead to the brake pipe; *g* leads to chamber D; EX, is the exhaust opening leading out at the back of the valve; *e* is the preliminary exhaust port leading to chamber D; *r* is the warning port leading to the exhaust; *p* is the port leading to the pump governor; *l* leads to the distributing valve release pipe; *u* leads to the application cylinder pipe.

Running Position.—This is the proper position of the handle 1, when the brakes are charged and ready for use; 2, when the brakes are not being operated; and 3, to release the locomotive brakes.

In this position, cavity *f* in the rotary valve connects ports *b* and *d* in the valve seat, affording a large direct passage from the feed-valve pipe to the brake pipe, so that the latter will charge up as rapidly as the feed valve can supply the air, but cannot attain a pressure above that for which the feed valve is adjusted.

Cavity *k*, in the rotary valve connects ports *c* and *g*, in the valve seat, so that chamber D, and the equalizing reservoir charge uniformly with the brake pipe, keeping the pressure on the two sides of the equalizing piston equal.

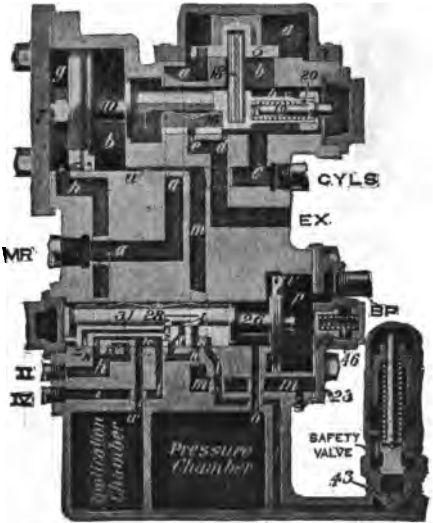


FIG. 1,986.—***Release, automatic*** or ***independent connections*** of distributing valve. It will be noted that main reservoir pressure is always present in the chamber surrounding application valve *5*, by its connection through passage *a*, *a*, to the main reservoir pipe. Chamber *b*, to the right of application piston *10*, is always in free communication with the brake cylinders, through passage *c*, and the brake cylinder pipe. Application cylinder *g*, at the left of application piston *10*, is connected by passage *h* with the equalizing valve seat, and to the brake valves through the application cylinder pipe. ***Charging.*** It will be seen that as chamber *p*, is connected to the brake pipe, brake pipe air flows through the feed groove *v*, over the top of piston *26*, into the chamber above equalizing valve *31*, and through port *o*, to the pressure chamber, until the pressures on both sides the piston are equal. ***Automatic release.*** When the automatic brake valve handle is placed in *release* position, and the brake pipe pressure in chamber *p*, is thereby increased above that in the pressure chamber, equalizing piston *26*, moves to the left, carrying with it equalizing valve *31*, and graduating valve *28*, to the position here shown. The feed groove *v*, now being open permits the pressure in the chamber to feed up until it is equal to that in the brake pipe as before described. This action does not release the locomotive brakes because it does not discharge application cylinder pressure. The release pipe is closed by the rotary valve of the automatic brake valve, and the application cylinder pipe is closed by the rotary valves of both brake valves. ***To release*** the locomotive brakes, the automatic brake valve must be moved to *running* position. The release pipe is then connected by the rotary valve to the atmosphere, and as exhaust cavity *k*, in the equalizing valve *31*, connects ports *i*, *w* and *h* in the valve seat, the air in the application cylinder and chamber will escape. As this pressure reduces, the brake cylinder pressure will force application piston *10*, to the left until exhaust valve *16*, uncovers exhaust ports *d* and *e*, allowing brake cylinder pressure to escape, or in case of graduated release, to reduce in like amount to the reduction in the application cylinder pressure.

Port *s*, in the rotary valve registers with port *p*, in the valve seat, permitting air at main reservoir pressure, which is present at all times above the rotary valve, to pass to the lower connection of the excess pressure head of the compressor governor.

Port *h*, in the rotary valve registers with port *l*, in the seat, connecting the distributing valve release pipe through the exhaust cavity EX, with the atmosphere.

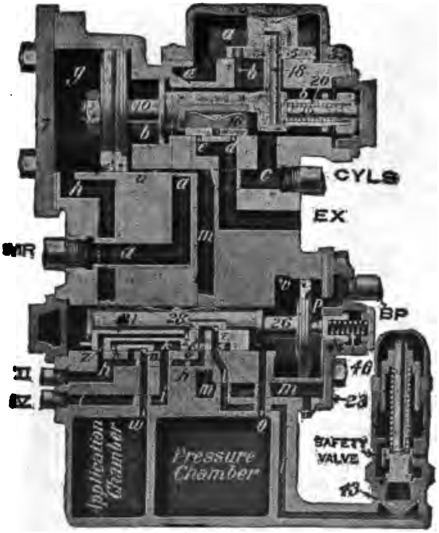


FIG. 1,987.—***Automatic service.*** When a service application is made with the automatic brake valve, the brake pipe pressure in chamber *p*, is reduced, causing a difference inpressure on the two sides of this piston, which results in the piston moving toward the right. The first movement of the piston closes the feed groove, and at the same time moves the graduating valve until it uncovers the upper end of port *z*, in the equalizing valve *31*. As the piston continues its movement, the shoulder on the end of its stem engages the equalizing valve which is then also moved to the right until the piston strikes equalizing piston graduating sleeve *44*, graduating spring *46*, preventing further movement; port *z*, in the equalizing valve then registers with port *h*, in the seat, and cavity *n*, in the equalizing valve connects ports *h*, and *w*, in the seat. As the equalizing valve chamber is always in communication with the pressure chamber, air can now flow from the latter to both the application cylinder and application chamber. This pressure forces application piston *10*, to the right, as shown in the figure causing exhaust valve *16*, to close exhaust ports *e*, and *d*, and to compress application piston graduating spring *20;* also causing application valve *5*, by its connection with the piston stem through pin *18*, to open its port and allow air from the main reservoirs to flow, into chambers *b*, *b*, and through passage *c*, to the brake cylinders. During the movement just described, cavity *t*, in the graduating valve connects ports *r*, and *s*, in the equalizing valve and by the same movement ports *r*, and *s*, are brought to register with ports *h*, and *l*, in the seat. This establishes communication between the application cylinder and the safety valve, which being set at 68 pounds (three pounds above the maximum obtained in an emergency application from 70 pounds brake pipe pressure) limits the brake cylinder pressure to this amount. The amount of pressure resulting in the application cylinder for a certain brake pipe service reduction, depends on the comparative volumes of the pressure chamber, application cylinder and its chamber. These volumes are such that with 70, pounds in the pressure chamber and nothing in the application cylinder and chamber, if they are allowed to remain connected by the ports in the equalizing valve, they will equalize at about 50 pounds.

If the brake valve be in running position when uncharged cars are cut in, or if, after a heavy brake application and release, the handle of the automatic brake valve be returned to *running* position too soon, the governor will stop the compressors until the difference between the hands on gauge No. 1 is less than 20 pounds. The compressors stopping from this cause, call the engineer's attention to the seriously wrong operation on his part, as running position results in delay in charging, and is liable to cause some brakes to stick. Release position should be used until all brakes are released and nearly charged.

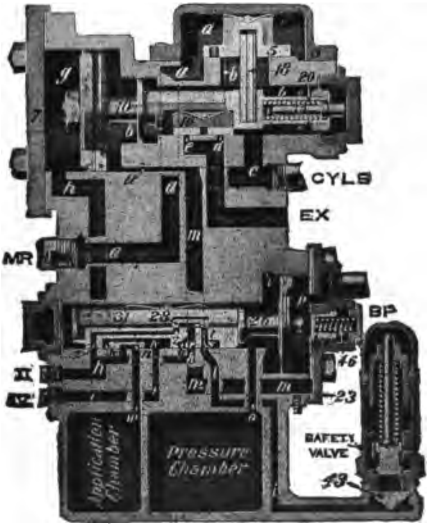


Fig. 1,988—*Service lap.* When the brake pipe reduction is not sufficient to cause a full service application, the conditions described above continue until the pressure in the pressure chamber is reduced enough below that in the brake pipe to cause piston *26*, to force graduating valve *28*, to the left until stopped by the shoulder on the piston stem striking the right hand end of equalizing valve *31*, the position here indicated, and known as *service lap*. ***In this position,*** graduating valve *28*, has closed port *z*, so that no more air can flow from the pressure chamber to the application cylinder and chamber. It also has closed port *s*, cutting off communication to the safety valve, so that any possible leak in the latter cannot reduce the application cylinder pressure, and thus similarly affect the pressure in the brake cylinders. The flow of air past application valve *5*, to the brake cylinders continues until their pressure slightly exceeds that in the application cylinder when the higher pressure and application piston graduating spring together force piston *10*, to the left to the position here shown, thereby closing port *b*. Further movement is prevented by the resistance of exhaust valve *16*, and the application piston graduating spring having expanded to its normal position. The brake cylinder pressure is then practically the same as that in the application cylinder and chamber. From the above description it will be seen that application piston *10*, has application cylinder pressure on one side *g*. and brake cylinder pressure on the other. When either pressure varies, the piston will move toward the lower. Consequently if that in chamber *b*, be reduced, by brake cylinder leakage. the pressure maintained in the application cylinder *g*, will force piston *10*, to the right, opening application valve *5*, and again admitting air from the main reservoir to the brake cylinders until the pressure in chamber *b*, is again slightly above that in the application cylinder *g*, when the position again moves back to lap position. In this way the brake cylinder pressure is always maintained equal with that in the application cylinder. This is the pressure maintaining feature.

Service Position.—This position gives a gradual reduction of brake pipe pressure to cause a service application.

Port *h*, in the rotary valve registers with port *c*, in the valve seat, allowing air from chamber D, and the equalizing reservoir to escape to the atmosphere through cavities *o*, in the rotary valve and EX in the valve seat. Port *e*, is restricted so as to make the pressure in the equalizing reservoir and chamber D, fall gradually.

As all other ports are closed, the fall of pressure in chamber D, allows the brake pipe pressure under the equalizing piston to raise it, and unseat its valve, allowing brake pipe air to flow to the atmosphere gradually through the opening marked BP Ex. When the pressure in chamber D, is reduced the desired amount, the handle is moved to lap position, thus stopping any further reduction in that chamber.

Air will continue to discharge from the brake pipe until its pressure has fallen to an amount a trifle less than that retained in chamber D, permitting the pressure in this chamber to force the piston downward gradually and stop the discharge of brake pipe air. It will be seen therefore, that the amount of reduction in the equalizing reservoir determines that in the brake pipe, regardless of the length of the train.

The gradual reduction of brake pipe pressure is to prevent quick action, and the gradual stopping of this discharge is to prevent the pressure at the head end of the brake pipe being built up by the air flowing from the rear, which might cause some of the head brakes to "kick-off."

Lap Position.—This position is used while holding the brakes applied after a service application until it is desired either to make a further brake pipe reduction or to release them. All ports are closed.

Release position.—This position, which is used for releasing the train brakes after an application, without releasing the locomotive brakes, has already been described under charging and release on page 1,009.

The air flowing from the main reservoir pipe connection through port *a*, in the rotary valve and port *b*, in the valve seat to the brake pipe, raises the pressure in the latter, thereby causing the triple valves and equalizing portion of the distributing valve to go to release position, which releases the train brakes and recharges the auxiliary reservoirs and the pressure chamber in the distributing valve.

When the brake pipe pressure has been increased sufficiently to cause this, the handle of the brake valve should be moved to either running or holding position; the former when it is desired to release the locomotive brakes, and the latter when they are to be still held applied.

Holding position.—This position is so named because the locomotive brakes are held applied while the train brakes are being released and their auxiliary reservoirs recharged to feed valve pressure. All ports register as in running position, except port *l*, which is closed.

Therefore, the only difference between running and holding positions is that in the former the locomotive brakes are released, while in the latter they are held applied.

Emergency Position.—This position is used 1, when the most prompt and heavy application of the brakes is required, and 2, to prevent loss of main reservoir air and insure that the brakes remain applied in the event of a burst hose, a break in two, or the opening of a conductor's valve.

Port *x*, in the rotary valve registers with port *c*, in the valve seat, making a large and direct communication between the brake pipe and atmosphere through cavity *o*, in the rotary valve and EX in the valve seat. This direct passage makes a sudden and heavy discharge of brake pipe air, causing the triple valves and distributing valve to move to emergency position and give maximum braking power in the shortest possible time.

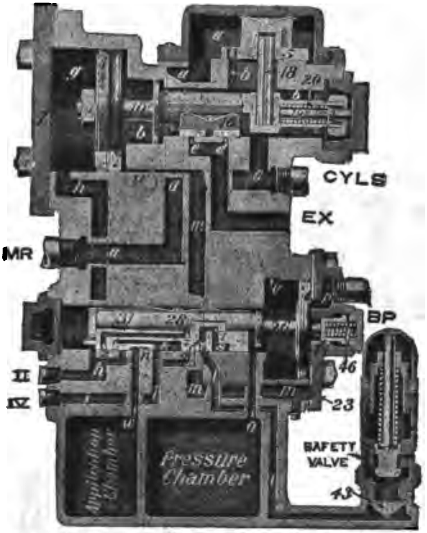


FIG. 1,989.—***Emergency.*** When a sudden and heavy brake pipe reduction is made, as in an emergency application, the air pressure in the pressure chamber forces equalizing piston *26*, to the right with sufficient force to compress equalizing piston graduating spring *46*, and to seat against the leather gasket beneath cap *23*. This movement causes equalizing valve *31*, to uncover port *h* in the seat without opening port *w*, making a direct opening from the pressure chamber to the application cylinder only, so that they quickly become equalized. This cylinder volume, being small, and connected with that of the pressure chamber at 70 pounds pressure, equalizes at about 65 pounds. Also in this position of the automatic brake valve, a small port in the rotary valve allows air from the main reservoirs to feed into the application cylinder pipe, and thus to the application chamber. The application cylinder is now connected to the safety valve through port *h*, in the seat, cavity *q*, and port *r*, in the equalizing valve, and port *l*, in the seat. Cavity *q*, and port *r*, in the equalizing valve are connected by a small port, the size of which permits the air in the application cylinder to escape through the safety valve at the same rate that the air from the main reservoirs, feeding through the rotary valve of the automatic brake valve, can supply it, preventing the pressure rising above the adjustment of the safety valve. In high speed brake service the feed valve is regulated for 110 pounds brake pipe pressure instead of 70, and main, reservoir pressure is 130 or 140 pounds. Under these conditions an emergency application raises the application cylinder pressure to about 93 pounds, but the passage between cavity *q*, and port *r*, is so small that the flow of application cylinder pressure to the safety valve is just enough greater than the supply through the brake valve, to decrease that pressure in practically the same time and manner as is done by the high speed reducing valve, until it is approximately 75 pounds. The reason why the pressure in the application cylinder pressure chamber and brake cylinders does not fall to 68 pounds to which pressure the safety valve is adjusted, is because the inflow of air through the brake valve with the high main reservoir pressure used in high speed service is equal, at 75 pounds, to the outflow through the small opening to the safety valve. This is done to get a shorter stop in emergency. The application portion of the distributing valve operates similarly, but more quickly than in service application.

In this position, main reservoir air flows to the application cylinder through port *j*, which registers with a groove in the seat connecting with cavity *k*, thence through ports *n*, in the valve and *u*, in the seat to the application cylinder pipe, thereby maintaining application cylinder pressure as described on page 1,013.

At the same time port *t*, (fig. 1,985), in the rotary valve registers with port *g*, in the seat, allowing the air in the equalizing reservoir to flow through the ports named to the exhaust *o*, and atmosphere, thus reducing the pressure in the equalizing reservoir to zero during an emergency application of the brakes.

Leather washer *8*, prevents air in the rotary valve chamber leaking past the rotary valve key to the atmosphere. Spring *30*, keeps the rotary valve key firmly

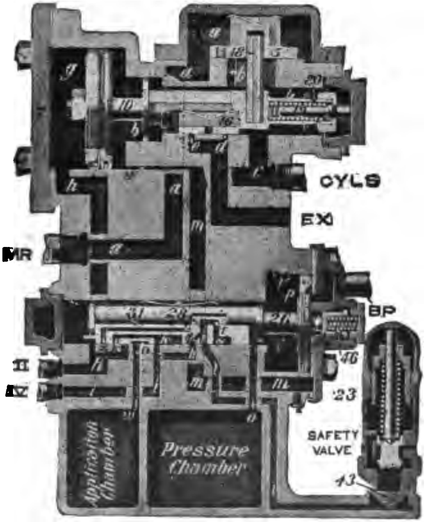


FIG. 1,990.—***Emergency Lap.*** The movable parts of the valve remain in the position shown in fig. 1,989 until the brake cylinder pressure slightly exceeds the application cylinder pressure, when the application piston and application valve move back to the position known as *emergency lap* as shown above. The release after an emergency is brought about by the same manipulation of the automatic brake valve as that following service application, but the effect on the distributing valve is somewhat different. When the equalizing piston, equalizing valve, and graduating valve are forced to the release position by the increased brake pipe pressure in chamber *p*, the application chamber, pressure in which is zero, is connected to the application cylinder, having emergency pressure therein through port *w*, cavity *k*, and port *h*. The pressure in the application cylinder at once expands into the application chamber until these pressures are equal, which results in the release of brake cylinder pressure until it is slightly less than that in application cylinder and chamber. Consequently, in releasing after an emergency (using the release position, of the automatic brake valve), the brake cylinder pressure will automatically reduce to about 15 pounds, where it will remain until the automatic brake valve handle is moved to running position. If the brakes be applied by a conductor's valve, a burst hose, or parting of train, the movement of equalizing valve *31*, breaks the connection between ports *h* and *i*, through cavity *k*, so that the brakes will apply and remain applied until the brake pipe pressure is restored The handle of the automatic brake valve should be immediately moved to either emergency or lap position. depending upon whether the locomotive of operating a passenger or freight train, as explained on page 1,033, to prevent a loss of main reservoir pressure.

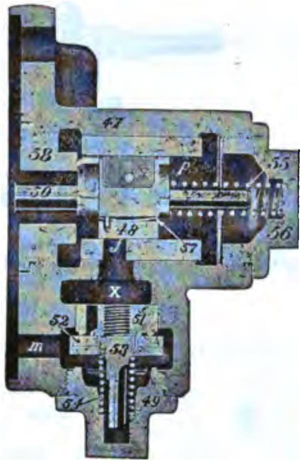


FIG. 1,991.—Distributing valve quick action cylinder cap. The equalizing portion of the distributing valve, as already described, corresponds to the plain triple valve of the old standard locomotive brake equipments. There are, however, conditions under which it is advisable to have it correspond to a quick action triple valve; that is, —vent brake pipe air into the brake cylinders in an emergency application. To obtain this, the cylinder cap *23*, fig. 1,980, is replaced by the "quick action cylinder cap," here illustrated. ***The parts are;*** *47*, cylinder cap; *48*, emergency valve; *49*, check valve cap nut; *50*, emergency valve graduating stem; *51*, check valve guide; *52*, rubber seat for check valve; *53*, check valve; *54*, check valve spring; *55*, graduating spring; *56*, cap nut; *57*, emergency valve spring; *58* stop plug. The quick action cylinder cap is recommended for double heading freight service and particularly for all passenger locomotives.

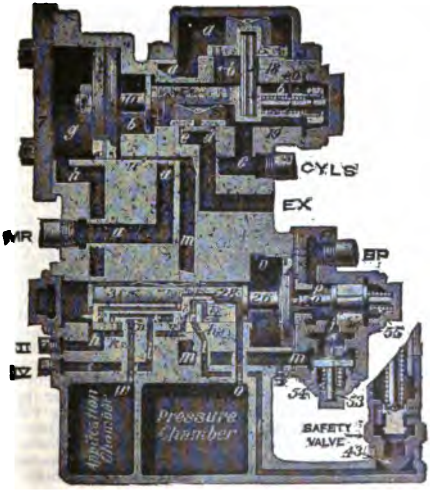
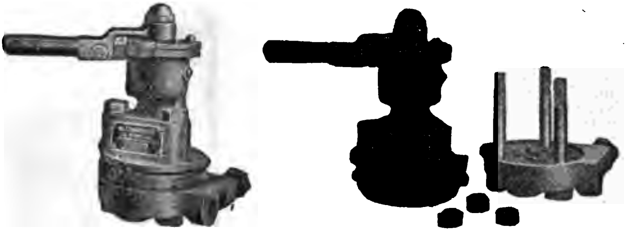
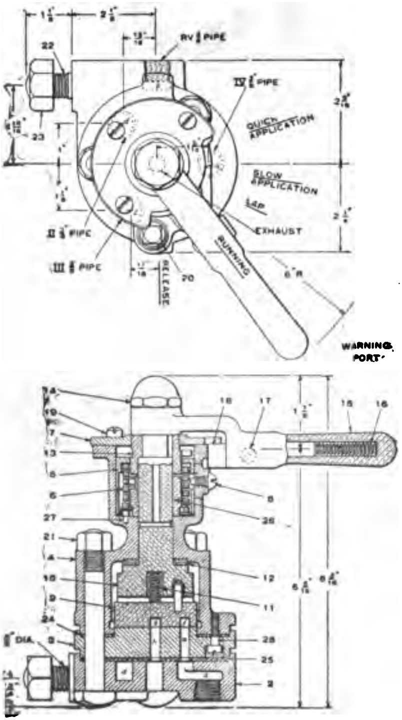


FIG. 1,992.—***Emergency position*** for distributing valve with quick action cap. In this position as equalizing piston *26*, moves to the right and seals against the gasket, the knob on the piston strikes the graduating stem *50*, causing it to compress equalizing piston graduating spring *55*, and move emergency valve *48*, to the right, opening port *j*. Brake pipe pressure in chamber *p*, flows to chamber *x*, pushes down check valve *53*, and passes to the brake cylinders through port *m* in the cap and distributing valve body. When the brake cylinders and brake pipe equalize, check valve *53*, is forced to its seat by spring *54*, thus preventing air in the brake cylinders flowing back into the brake pipe. When a release of the brakes occurs and piston *26* is moved back to its normal position fig. 1,986, spring *55*, forces graduating stem *50*, and emergency valve *48*, back to the position shown in fig. 1,991. In all other respects, the operation of a distributing valve having this cap is exactly as described before.



Figs. 1,993 to 1,998—Independent brake valve. Fig 1,993, valve complete; figs. 1,994 to 1,998, valve removed from pipe bracket.



Figs. 1,999 and 2,000.—Plan and elevation of independent brake valve. ***Connections:*** RV, reducing valve pipe; EX, exhaust;IV, distributing valve release pipe to the distributing valve; 111, distributing valve, release pipe to automatic brake valve; 11, application cylinder pipe. ***Parts:*** 2, pipe bracket; 3, rotary valve seat; 4, valve body; 5, return spring casing; 6, return spring; 7, cover; 8, casing screw; 9, rotary valve; 10, rotary valve key; 11, rotary valve spring; 12, key washer; 13, upper clutch; 14, handle nut; 15, handle; 16, latch spring; 17, latch screw; 18, latch; 19, cover screw; 20, oil plug; 21, bolt and nut; 22, bracket stud; 23, bracket stud nut; 24, upper gasket; 25, lower gasket 26, lower clutch; 27, return springs top; 28, cap screw. Port *b*, in the seat leads to the reducing valve pipe. Port *a*, leads to that portion of the distributing valve release pipe which connects to the distributing valve at IV (fig. 1,986). Port *c*, leads to the other portion of the release pipe which connects to the automatic brake valve at III (fig. 1,981). Port *d*, leads to the application cylinder pipe which connects to the distributing valve at II, (fig. 1,986). Port *h*, in the center, is the exhaust port leading directly down to the atmosphere. Port *k*, is the warning port, connecting with the atmosphere.

pressed against washer 8, when no main reservoir pressure is present. The handle 9 contains latch 11, which fits into notches in the quadrant of the top case, so located as to indicate the different positions of the brake valve handle. Handle latch spring 10, forces the latch against the quadrant with sufficient pressure to indicate each position.

To remove the brake valve, close the cocks, and take off the holding down nuts. To take the valve proper apart, remove the cap screws.

The brake valve should be located so that the engineer can operate it conveniently from his usual position, while looking forward or back out of the side cab window.

Independent Brake Valve.—Figs. 1,993 to 1,998 show this valve, which is of the rotary type. Figs. 1,999 and 2,000 show sectional views of the valve. In these views, the pipe connections and positions of the handle are indicated. The five positions of the brake valve handle are, beginning at the extreme left, release, running, lap, slow application and quick application.

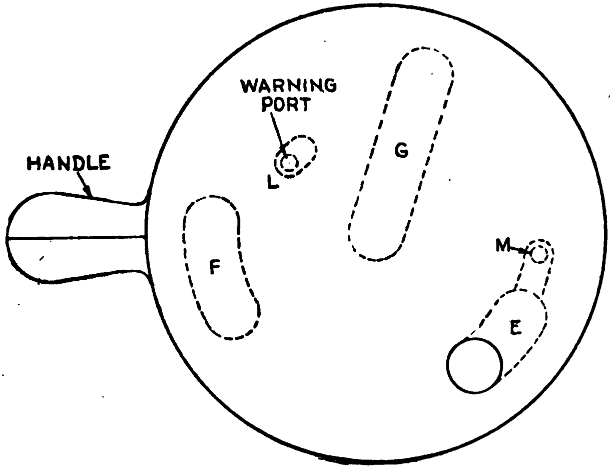


Fig. 2,001.—Rotary valve of Westinghouse S–6 independent brake valve. The exhaust cavity G, is always in communication at one end with exhaust port H. Groove E, in the face of the valve communicates at one end with a port through the valve. This groove is always in communication with a groove in the seat connecting with supply port B, and through the opening just mentioned, air is admitted to the chamber above the rotary valve, thus keeping it to its seat. Port M, connects by a small hole with groove E; F, is a groove in the face of the rotary valve; L, is the warning port, extending through the rotary valve.

Figs. 2,002 and 2,003 are top views of both brake valves showing the position of the handles.

The various positions of the independent valve are as follows:

Running Position.—This is the position that the independent brake valve should be carried in at all times when the independent brake is not in use.

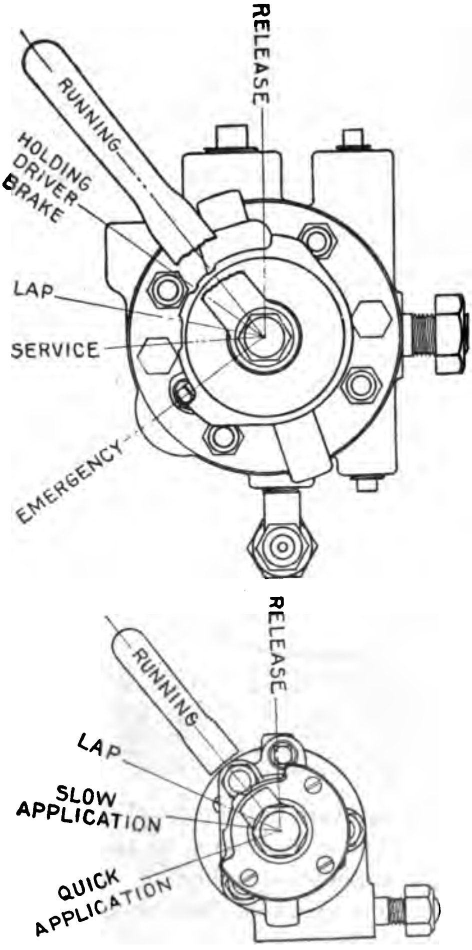
Groove *f*, in the rotary valve connects ports *a*, and *c*, in the valve seat, thus establishing communication through the distributing valve release pipe, between the application cylinder of the distributing valve and port *l*, of the automatic brake valve figs. 1,981 and 1,982, so that the distributing valve can be released by the latter.

It will also be noted that if the automatic brake valve be in running position, and the independent brakes are being operated, they can be released by simply returning the independent valve to running position, as the application cylinder pressure can then escape through the release pipe and automatic brake valve.

Slow Application Position.—To apply the independent brake lightly or gradually, move the brake valve handle to the slow application position.

Port *m*, registers with port *d*, allowing air to flow from the reducing valve pipe through port and groove *b*, in the seat, groove *e*, in the rotary valve, and the comparatively small port *m*, to port *d*; thence through the application cylinder pipe to the application cylinder of the distributing valve.

Quick Application Position.—To obtain a quick application of the independent brake, move the brake valve handle to quick application position, groove *e*, then connects ports *b*, and *d*, directly, making a larger opening between them in the slow application position, allowing supply air to flow rapidly from the reducing valve pipe to the application cylinder of the distributing valve.



FIGS. 2,002 and 2,003.—Positions of automatic and independent brake valve handles.

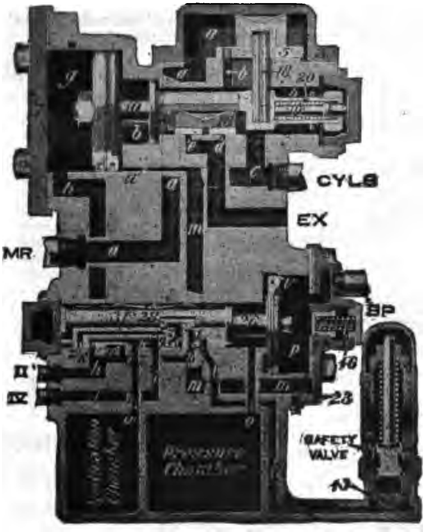


FIG. 2,004—***Independent application.*** When the handle of the independent brake valve is moved to either *slow* or *quick* application position, air from the main reservoir, limited by the reducing valve to a maximum of 45 pounds, is allowed to flow to the application cylinder, forcing application piston *10*, to the right as here shown. This movement causes application valve *5*, to open its port and allow air from the main reservoirs to flow into chambers *b*,*b*, and through passage *c*, to the brake cylinders, as in an automatic application, until the pressure slightly exceeds that in the application cylinder. The application piston graduating spring *20*, and higher pressure then force application piston *10*, to the left until application valve *5*, closes its port. Further movement is prevented by the resistance of exhaust valve *16*, and the application piston graduating spring having expanded to its normal position. This position shown in fig. 2,005, is known as ***independent lap.***

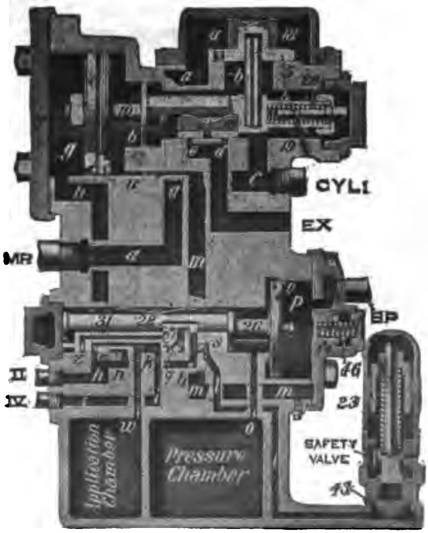


FIG. 2,005.—***Independent lap,*** as described under fig. 2,004,.

Since the supply pressure to the valve is fixed by the regulation of the reducing valve to 45 pounds, this is the maximum cylinder pressure that can be obtained.

Lap Position.—This position is used to hold the independent brake applied after the desired cylinder pressure is obtained, at which time all communication between operating ports is closed.

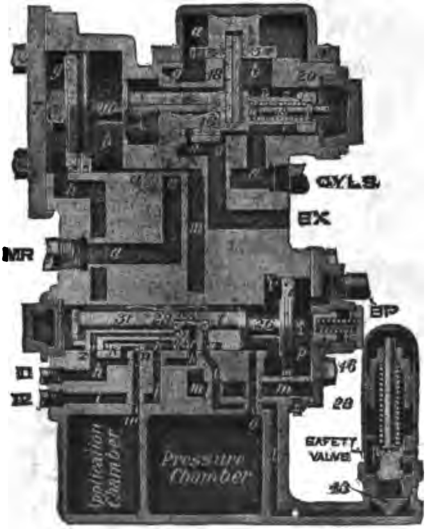


Fig. 2,006.—*Independent release.* When the handle of the independent brake valve is moved to release position, a direct opening is made from the application cylinder to the atmosphere. As the application cylinder pressure escapes, brake cylinder pressure in chamber *b*, moves application piston *10*, to the left, causing exhaust valve *16*, to open exhaust ports *e* and *d*, as shown in fig. 1,986, thereby allowing brake cylinder pressure to discharge to the atmosphere. If the independent brake valve be returned to lap before all of the application cylinder pressure has escaped, the application piston *10*, will return to *independent lap* position, (fig. 2,005), as soon as the brake cylinder pressure is reduced a little below that remaining in the application cylinder, thus closing exhaust ports *e* and *d*, and holding the remaining pressure in the brake cylinders. In this way the independent release may be graduated as desired. The above figure shows the position the distributing valve parts will assume, if the locomotive brakes be released by the independent brake valve after an automatic application has been made. This results in the application portion going to release position without changing the conditions in either the pressure chamber or brake pipe; consequently, the equalizing portion does not move until release is made by the automatic brake valve. An independent release of locomotive brakes may also be made in the same manner, after an emergency application by the automatic brake valve. However, owing to the fact that, in this position, the automatic brake valve will be supplying the application cylinder through the maintaining port in the rotary valve (see fig. 1,989), the handle of the independent brake valve must be held in release position to prevent the locomotive brakes reapplying, so long as the handle of the automatic brake valve remains in emergency position. The equalizing portion of the distributing valve will remain in the position shown in figs. 1,989 and 1,990, while the application portion will assume the position here shown.

Release Position.—This position is used to release the pressure from the application cylinder when the automatic brake valve is not in running position.

At such time, the cavity *g*, fig. 2,000, registers with port *d*, allowing the air in the application cylinder to flow through the application cylinder pipe, ports *d*, *g* and *h*, to the atmosphere.

The purpose of return spring *6*, is to automatically move the handle *15*, from the release to the running position or from the quick application to the slow application position, as soon as the engineer lets go of it. The automatic return from release to running position is to prevent leaving the handle in release position, and thereby make it impossible to operate the locomotive brake with the automatic brake valve. The action of the spring between quick application and slow application position serves to make the latter more prominent, so that in rapid movement of the valve, the engineer is less likely to unintentionally pass over to the quick application position, thereby obtaining a heavy application of the locomotive brake when only a light one was desired.

As a warning to the engineer in case of a broken return spring, port *l*, in the face of the rotary registers in release position with port *k*, in the seat, allowing air to escape to the atmosphere.

Figs. 2,002 and 2,003, show a top view of both brake valves, showing the position of their handles.

Double Heading.—When there are two or more locomotives in a train, the instructions already given remain unchanged so far as the leading locomotive, or the locomotive from which the brakes are being operated, is concerned. On all other locomotives in the train, however, the double heading cock under the automatic brake valve must be closed and the automatic and independent brake valve handles carried in running position. The release pipe is then open to the atmosphere at the automatic brake valve, and the operation of the distributing valve is the same as that described during automatic brake applications. In double heading, therefore, the application and the release of the distributing valve on each helper locomotive is similar to that of the triple valves on the train. But in case an engineer or a helper find it necessary to apply or to release his brakes independently of the train, he can do so by using the independent brake valve, without moving the handle of the automatic brake valve.

Feed Valve.—The latest feed valve differs from previous ones in charging to the regulated pressure somewhat quicker, and in

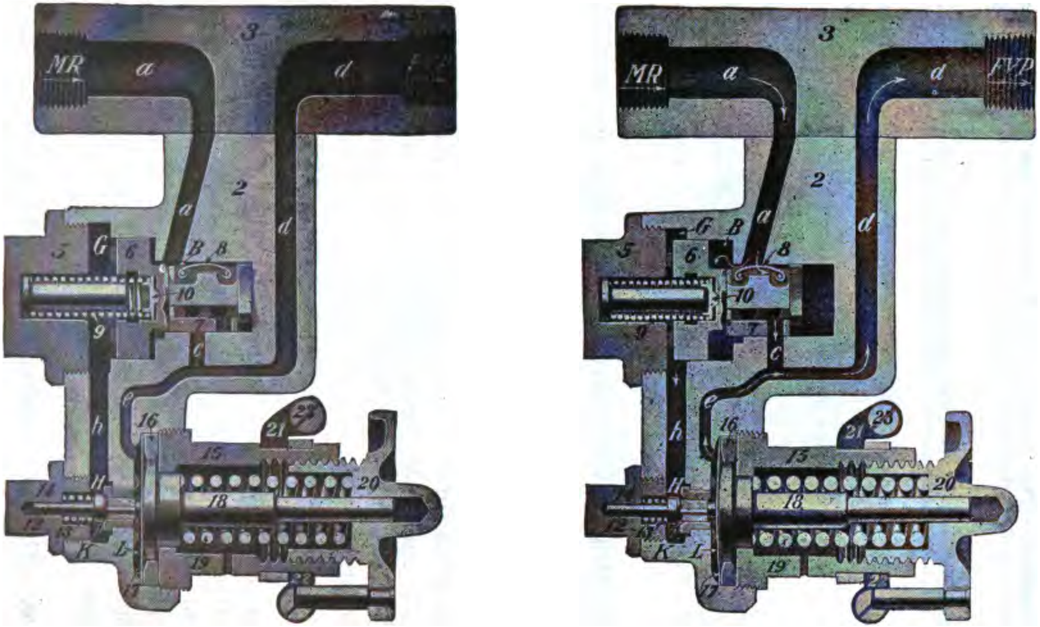


FIG. 2,007 and 2,008.—Westinghouse B-6 feed valve. Fig. 2,007 closed position; fig. 2,008, open position. ***Connections;*** M R, main reservoir pipe; FVP, feed valve pipe. ***Parts;*** *2*, valve body; *3*, pipe bracket; *5*, cap nut; *6*, supply valve piston, *7*, supply valve; *8*, supply valve spring; *9*, piston spring; *10*, piston spring tip; *12*, regulating valve; *13*, regulating valve spring; *14*, regulating valve cap; *15*, spring box; *16*, diaphragm ring; *17*, diaphragm; *18*, diaphragm spindle; *19*, regulating spring; *20*, regulating hand wheel; *21*, upper stop; *22*, lower stop; *23*, stop screw. ***The distinguishing feature*** of this type of feed valve is the duplex adjusting arrangement by which it eliminates the necessity of the two feed valves in high and low pressure service. The spring box, *15*, has two rings encircling it, which are split through the lugs marked *21* and *22*, in the diagram, and which may be secured in any position by the screw *23*. The pin forming part of adjusting handle *20*, limits the movement of the maintaining the pressure more accurately under the variable conditions of short and long trains, and of good and poor maintenance. Also, it gives high and low brake pipe pressure control. It is supplied with air directly from the main reservoir. It regulates the pressure in the feed valve pipe, and in the brake pipe when the handle of the automatic brake valve is in running or in holding positions, these two pipes being then connected through the brake valve. It is connected to a pipe bracket located in the piping between the main reservoir and the automatic brake valve, and is interchangeable with previous types.

Figs. 2,007 and 2,008 are diagrammatic views of the valve and pipe bracket having the ports and operating parts in one plane to facilitate description.

This feed valve consists of two sets of parts, the supply and regulating. The supply parts, which control the flow of air through the valve, consist of the supply valve *7*, and its spring *8*; the supply valve piston *6*, and its spring *9*. The regulating parts consist of the regulating valve *12*, regulating valve spring *13*, diaphragm *17*, diaphragm spindle *18*, regulating spring *19*, and regulating hand wheel *20*.

Main reservoir air enters through port, *a, a,* to the supply valve chamber *B*, forces supply valve piston *6*, to the left, compresses piston spring *9*, and causes the port in supply valve *7*, to register with port *c*, as in fig. 1,999. This permits air to pass through ports *c* and *d*, to the feed valve pipe at *FVP*, and through port *e*, to diaphragm chamber *L*.

Regulating valve *12*, is then open and port *K*, connects chamber *G*, on the left of piston *6*, to the feed valve pipe through passage *h*. chamber *L*, and passage *e, d, d.* While the regulating valve is open, air feeding by the piston *6*, cannot accumulate in chamber *G*, above feed valve pipe pressure, but when regulating valve *12*, is closed, the pressure on the left of piston *6*, quickly rises to the main reservoir pressure on the right and piston spring

FIGS. 2,007 and 2,008.—*Text continued.*

handle to the distance between stops *21* and *22*. When testing the valve, stop *21*, is located so that the compression of spring *19*, will give the desired high brake pipe pressure, and stop *22*, so that the spring compression is enough less to give the low brake pipe pressure. Thereafter, by simply turning handle *20*, until its pin strikes either one of these stops, the regulation of the feed valve is changed from one brake pipe pressure to the other. ***To adjust this valve***, slacken screw *23*, which allows stops *21* and *22*, to turn around spring box *15*. Adjusting handle *20*, should be turned until the valve closes at the lower brake pipe pressure desired, when stop *22*, should be brought into contact with the handle pin, at which point it should be securely fastened by tightening screw *23*. Adjusting handle *20*, should then be turned until the higher adjustment is obtained, when stop *21*, is brought in contact with the handle pin and securely fastened. It is recommended that the stops be placed to give 110 pounds high, and 70 pounds low, brake pipe pressure. When replacing this feed valve on its pipe bracket after removal, the gasket must always be in place between the valve and bracket, to insure a tight joint.

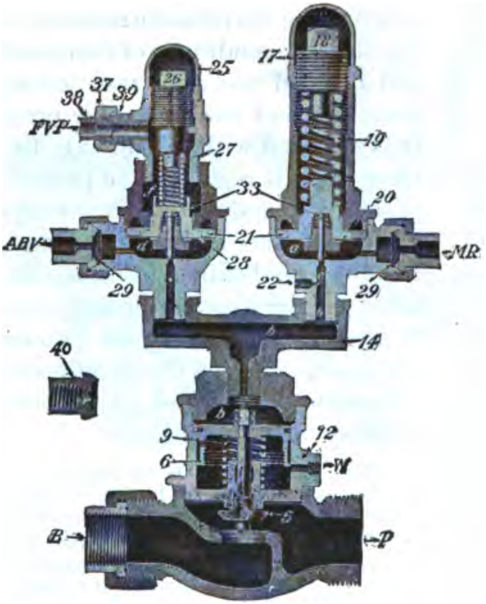


Fig. 2,009 and 2,010.—Westinghouse type SF-4 compressor governor, with steam valve ***5***, open. By reference to the piping diagram in fig. 1,936, it will be seen that connection ***B***, leads to the boiler; ***P***, to the air compressor; ***MR***, to the main reservoir; ***ABV***, to the automatic brake valve; ***FVP***, to the feed valve pipe; ***W***, is the waste pipe connection. Steam enters at ***B***, and passes by steam valve ***5***, to the connection P, and to the compressor. The governor regulating head on the left is called the "excess pressure head," and the one on the right the "maximum pressure head." Air from the main reservoir flows through the automatic brake valve (when the latter is in release, running or holding position) to the connection marked ***ABV***, and into chamber ***d***, below diaphragm ***28***. Air from the feed valve pipe enters at the connection ***FVP***, to chamber ***f***, above diaphragm ***28***, adding to the pressure of regulating spring ***27***, in holding it down. As this spring is adjusted to about 20 pounds, this diaphragm will be held down until the main reservoir pressure in chamber ***d***, slightly exceeds the combined air and spring pressure acting on the top of the diaphragm. At such time diaphragm ***28***, will rise, unseat its pin valve, and allow air to flow to chamber ***c***, above the governor piston, forcing the latter downward, compressing its spring and restricting the flow of steam past steam valve ***5***, to the point where the compressor will just supply the leakage in the brake system. When main reservoir pressure in chamber ***d***, becomes reduced, the combined spring and air pressure above the diaphragm forces it down, seating its pin valve. As chamber ***b***, is always open to the atmosphere through the small vent port ***c***, the air in chamber ***b***, above the governor piston will then escape to the atmosphere and allow the piston spring, and steam pressure below valve ***5***, to raise it and the governor piston to the position shown. Since the connection from the main reservoir to chamber ***d***, is open only when the handle of the automatic brake valve is in release, *running* or *holding* positions, in the other positions this governor head is cut out. The connection marked ***MR***, in the maximum pressure head should be connected to the main reservoir cut out cock, or to the pipe connecting the two main reservoirs, so as to be always in communication with the main reservoir, so that when the excess pressure head is cut out by the brake valve, or by the main reservoir cut out

6, forces piston *8*, and supply valve *9*, to the right, closes port *c*, and stops the flow to the feed valve pipe.

The regulating valve is operated by diaphragm *14*. When the pressure of regulating spring *17*, on its right is greater than the feed valve pressure in chamber *L*, on its left, it holds regulating valve *12*, open. This causes the supply valve to admit air to the feed valve pipe. When the feed valve pipe pressure in chamber *L*, is greater than that of the regulating spring *17*, the diaphragm allows regulating valve *12*, to close. This causes the supply valve to stop admitting air to the feed valve pipe.

As explained on page 1,010 under H-6 automatic brake valve, in release position of the latter, the warning port is supplied from the feed valve pipe. This insures that the excess pressure governor head will regulate the brake pipe pressure in release position even though the feed valve be leaking slightly but not enough to be otherwise detrimental.

Compressor Governor.—The duty of the compressor governor is to sufficiently restrict the speed of the compressor, when the desired main reservoir pressure is obtained, to prevent this pressure rising any higher.

During most of the time on a trip, the automatic brake valve is in running position, keeping the brakes charged. But little excess pressure is then needed, and the governor regulates the main reservoir pressure to about 20 pounds only above the brake pipe pressure, thus making the work of the compressor easier. On the other hand, when the brakes are applied (lap position of the automatic brake valve, following the use of its service position) a high main reservoir pressure is needed to insure their prompt release and recharge. Therefore, as soon as the use of lap service or emergency positions is commenced, the governor allows the compressor to work freely until the maximum main reservoir pressure is obtained. Again, when the brake pipe pressure is changed from one amount to another by the feed valve, as where a locomotive is used alternately in high speed brake and ordinary service, the governor automatically changes the main reservoir pressure to suit, and at the same time maintains the other features just described.

Another feature is that the governor connections to the brake valve permit the engineer to raise and maintain the brake pipe pressure about

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FIGS. 2,009 and 2,010—*Text continued.*
cock, the maximum pressure head will control the compressor. When main reservoir pressure in chamber *a*, exceeds the adjustment of spring *19*, in the maximum pressure head, diaphragm *20*, will raise its pin valve and allow air to flow into chamber *b*, above the governor piston, controlling the compressor as above described. The adjustment of spring *19*, thus fixes the maximum limit of main reservoir pressure, during such time as the automatic brake valve handle is in lap, service or emergency positions. As each governor head has a vent port *c*, from which a small amount of air escapes whenever pressure is present in port *b*, to avoid an unnecessary waste of air, one of these should be plugged.

20 pounds above the feed valve regulation before commencing and during the descent of steep grades, merely by the use of release position of the automatic brake valve, the position which should be used during such braking. Fig. 2,009 is a sectional view showing governor mechanism. To adjust the excess pressure head of this governor, remove cap nut *25*, and turn adjusting nut *26*, until the compression of spring *27*, gives the desired difference between main reservoir and brake pipe pressures. While adjusting the excess pressure head, the handle of the automatic brake valve should be in running position. To adjust the maximum pressure head remove cap nut *17*, and turn adjusting nut *18*, until the compression of spring *19*, causes the compressor to stop at the maximum main reservoir pressure required. While adjusting the maximum pressure head the handle of the automatic brake valve should be on lap. It is recommended that spring *27*, be adjusted for 20 pounds excess pressure, and spring *19*, for a pressure ranging from 120 to 140 pounds, depending on the service.

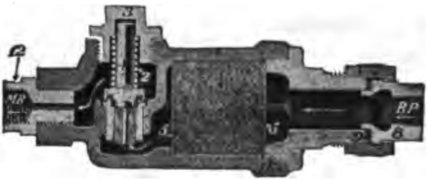


Fig. 2,011.—Westinghouse combined air, strainer and check valve. With the cut out cock open, air from the brake pipe enters at *BP*, passes through the curled hair strainer, lifts check valve *4*, held to its seat by a strong spring *2*, passes through the choke bushing, and out at *MR*, to the main reservoir, thus providing pressure for operating the brakes on the locomotive. The double heading cock should be closed, and the handle of each brake valve should be in running position. Where absence of water in the boiler, or other reason, justifies keeping the maximum braking power of such a locomotive lower than the standard, this can be accomplished by reducing the adjustment of the safety valve on the distributing valve. It can also be reduced at will by the independent brake valve. The strainer protects the check valve and choke from dirt. Spring *2*, over the check valve insures this valve seating and, while assuring an ample pressure to operate the locomotive brakes, keeps the main reservoir pressure somewhat lower than the brake pipe pressure, thereby reducing any leakage from the former. The choke prevents a sudden drop in brake pipe pressure and the application of the train brakes, which would otherwise occur with an uncharged main reservoir cut in to a charged brake pipe.

Dead Engine Device.—This feature as shown in fig. 1,936, is for the operation of the locomotive brake when the compressor on a locomotive in a train is inoperative from any cause. Fig. 2,011 shows the combined strainer, check valve, and choke. As these parts are not required at other times, a cut out cock is provided.

This cock should be kept closed except under the conditions just mentioned. The air for operating the brakes on such a locomotive must then be supplied through the brake pipe from the locomotive operating the train brakes.

Practical Points on Operation.—The following instructions are general, and must necessarily be supplemented to a limited extent to fully meet the varying local conditions on different railways. The instructions for operating the Westinghouse ET equipment are practically the same as for the combined automatic and straight air brake, hence no special departure from present methods of brake operation is required to get the desired result. Briefly the brakes should be operated as follows:

When not in use, carry the handles of both brake valves in running position.

To apply the brakes in service, move the handle of the automatic brake valve to the service position, making the required brake pipe reduction then back to lap position which is one for holding all the brakes applied.

To make a smooth and accurate two application passenger stop, make the first application sufficiently heavy to bring the speed of train down to about 15 miles per hour at a convenient distance from the stopping point, then release as explained in the following paragraph and re-apply as required to make the desired stop, the final release being made as explained below.

Releasing Brakes.—With the changes in operating conditions and in train and locomotive equipments during the past few years it has become possible to obtain still better results in general train handling if the method of operating the brakes be also slightly changed to conform with the progress which has been made in other directions.

This is especially true with regard to releasing brakes, and the general instructions which follow are intended to apply particularly to trains having modern equipment, that is, large compressor capacity, large main reservoir volume, high excess pressure and operating under present day average conditions. They are not intended to apply rigidly to all individual cases or conditions, specific instructions usually being issued by each road to cover its own recommended practice.

Passenger Service.—In making the first release of a two application stop, the brake valve handle should be moved to release position and then quickly back to running position, where it should be allowed to remain for an instant (1st, to permit the pressure in the equalizing reservoir and brake pipe to equalize and 2nd, to release part of the driver brake cylinder pressure), then moved to lap position and from there to service position as required. In passenger service the time the handle is in release position should be only momentary but the time in running position should be governed by the conditions existing for each particular case, such as the length of train, kind of reduction made, time available, and so on.

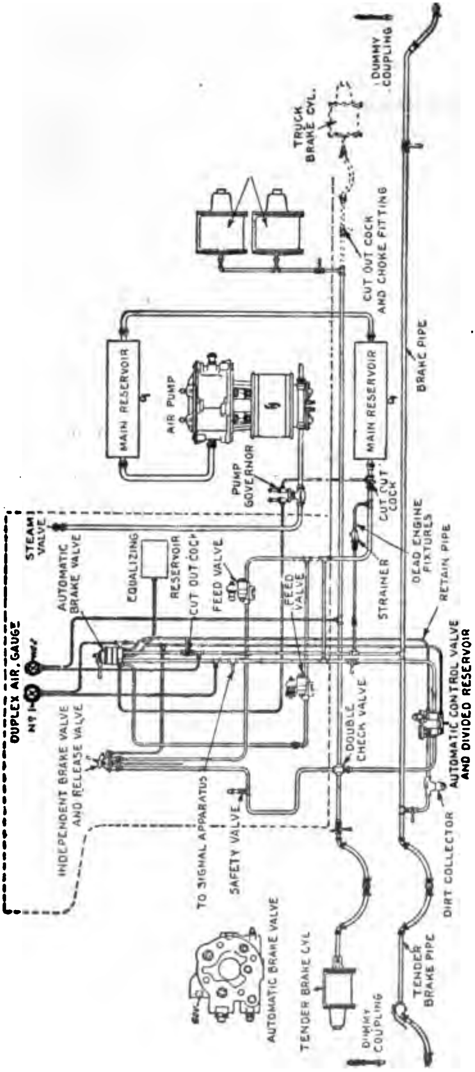


FIG. 2,012.—Piping diagram of New York LT-1 automatic control equipment.

In making the final release of a two application stop, with short trains, release shortly before coming to a standstill by moving the handle to release position and immediately back to running position, and leave it there. With long trains, the brakes should, as a rule, be held applied until the train stops.

The release after a one application stop, should be made in the same manner as the final release of a two application stop.

Freight Service.—Under present conditions it is, as a rule, safest to come to a stop before releasing the brakes on a freight train, especially a long one, rather than attempt to release at low speed. However, if conditions (for example, a short train, or a train equipped with type K, triple valves) permit of the release while in motion, the brake valve handle should be moved to release position and held there long enough to move as many of the triple valves to release position as possible without unduly overcharging the head end of the train (the time in release position should be governed by the length of train, amount of reduction made, etc.) then returned to running position to release the locomotive brakes and complete the recharging of the auxiliary reservoirs.

A few seconds after such a release, particularly on long trains, it is necessary to again move the handle to release position and quickly back to running position to "kick off" any brakes at the head end of the train that may have re-applied due to their auxiliary reservoirs having been slightly overcharged.

Holding Locomotive Brakes Applied.—If, when releasing as explained above, it be desired to hold the locomotive brakes applied after the other brakes release, move the handle from release back to holding instead of running position, then releasing the locomotive brakes fully by moving the handle to running position and leaving it there, or graduating them off, as circumstances require, by short, successive movements between holding and running positions.

A little experience with the ET equipment will enable the engineer to make smooth and accurate stops with much greater ease than was heretofore possible.

To apply the brakes in emergency, move the handle of the automatic brake valve quickly to emergency position and leave it there until the train stops and the danger is past.

When using the independent brake only, the handle of the automatic brake valve should be carried in running position. The independent application may be released by moving the independent brake valve handle to running position. Release position is for use only when the automatic brake valve handle is not in running position.

While handling long trains of cars, in road or switching service, the independent brake should be operated with care, to prevent damage to cars and lading, caused by running the slack in or out too hard. In cases of emergency arising while the independent brake is applied, apply the automatic brake instantly. The safety valve will restrict the brake cylinder pressure to the proper maximum.

The brakes on the locomotive and on the train may be alternated in heavy grade service where conditions (such as short, steep grades or where grade is heavy and straight for short distance) require, to prevent overheating of driving wheel tires and to assist the pressure retaining valves in holding the train while the auxiliary reservoirs are being recharged. This is done by keeping the locomotive brakes released by use of the independent brake valve when the train brakes are applied, and applying the locomotive brakes just before the train brakes are released, and then releasing the locomotive brakes after the train brakes are re-applied.

Care and judgment should always be exercised in the use of driver brakes on grades to prevent overheating of tires.

When all brakes are applied automatically, to graduate off or entirely release the locomotive brakes only, use release position of the independent brake valve.

The red hand of gauge No. 2 (fig. 1,963) will show at all times the pressure in the locomotive brake cylinders, and this hand should be watched in brake manipulation.

Release position of the independent brake valve will release the locomotive brakes under any and all conditions.

The train brakes should invariably be released before detaching the locomotive, holding with hand brakes where necessary. This is especially important on a grade, as there is otherwise no assurance that the car, cars, or train so detached will not start when the air brakes leak off, as they may in a short time where there is considerable leakage.

The automatic brakes should never be used to hold a locomotive or a train while standing even where the locomotive is not detached, for longer than ten minutes, and not for such time if the grade be very steep or the condition of the brakes be not good. The safest method is to hold with hand brakes only and keep the auxiliary reservoirs fully charged, so as to guard against a start from brakes leaking off, and to be ready to obtain any part of full braking power immediately on starting.

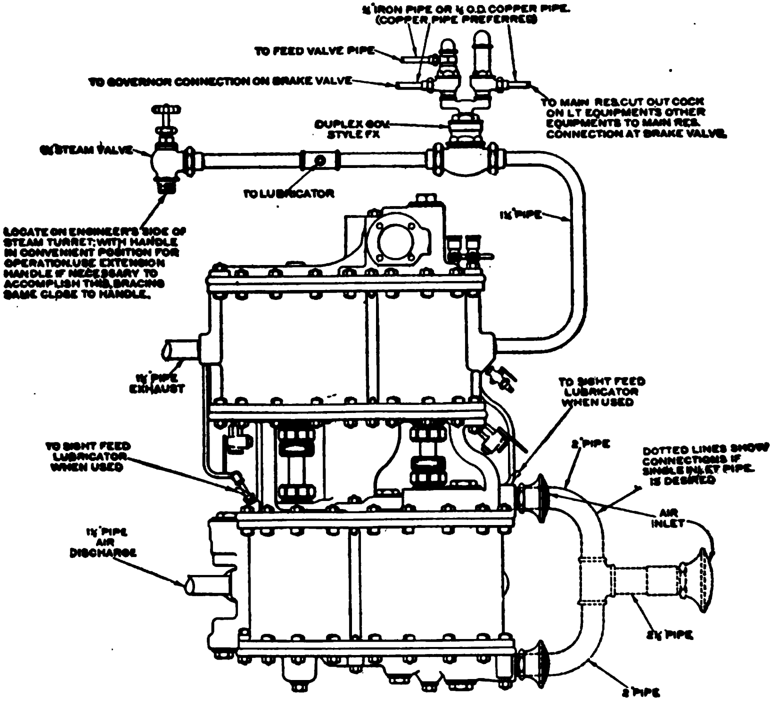

¼" IRON PIPE OR ¼ O.D. COPPER PIPE. (COPPER PIPE PREFERRED)
TO FEED VALVE PIPE
TO GOVERNOR CONNECTION ON BRAKE VALVE
TO MAIN RES. CUT OUT COCK ON LT EQUIPMENTS OTHER EQUIPMENTS TO MAIN RES. CONNECTION AT BRAKE VALVE.
DUPLEX GOV. STYLE FX
¾ STEAM VALVE
TO LUBRICATOR
1¼" PIPE
LOCATE ON ENGINEER'S SIDE OF STEAM TURRET; WITH HANDLE IN CONVENIENT POSITION FOR OPERATION. USE EXTENSION HANDLE IF NECESSARY TO ACCOMPLISH THIS, BRACING SAME CLOSE TO HANDLE.
1¼" PIPE EXHAUST
TO SIGHT FEED LUBRICATOR WHEN USED
TO SIGHT FEED LUBRICATOR WHEN USED
2" PIPE
DOTTED LINES SHOW CONNECTIONS IF SINGLE INLET PIPE IS DESIRED
AIR INLET
1¼" PIPE AIR DISCHARGE
2½" PIPE
2" PIPE

FIG. 2,013.—Installation diagram of New York 8½ inch compound air compressor.

The independent brake is a very important safety feature in this connection, as it will hold a locomotive with a leaky throttle or quite a heavy train on a fairly steep grade if, as the automatic brakes be released, the slack be prevented running in or out (depending on the tendency of the grade) and giving the locomotive a start. *To illustrate:*—the best method to make a stop on a descending grade is to apply the independent brake heavily as the stop is being completed, thus bunching the train solidly; then, when stopped, place and leave the handle of the independent brake valve in application position, then release the automatic brakes and keep them charged.

Should the independent brake be unable to prevent the train starting, the automatic brakes will become sufficiently recharged to make an immediate stop; in such an event enough hand brakes should at once be applied as are necessary to assist the independent brake to hold the train. Many runaways and some serious wrecks have resulted through failure to comply with the foregoing instructions.

When leaving the engine, while doing work about it, or when it is standing at a coal chute or water plug, always leave the independent brake valve handle in application position.

After an emergency application of the brakes while running over the road due to any cause other than intended by the operating engineer himself: 1, In passenger service move the brake valve handle to emergency position at once and leave it there until the train stops. 2, In freight service move the brake valve handle to lap position and let it remain there until the train stops.

This is to prevent loss of main reservoir pressure and insure the brakes remaining applied until released by the engineer in charge of the train. After the train stops the cause of the application should be located and remedied before proceeding.

Where there are two or more locomotives in a train, the instructions already given remain unchanged so far as the leading locomotive, or the locomotive from which the brakes are being operated, is concerned. On all other locomotives in the train, however, the double heading cock under the automatic brake valve must be closed and the automatic and independent brake valve handles carried in running position.

Before leaving the round house, the engineer should try the brakes with both brake valves, and see that no serious leaks exist. The pipes between the distributing valve and the brake valve should be absolutely tight.

Air Compressor.—The compressed air for the air brake is supplied by an air compressor, driven by steam, and located on the engine. The compressor consists primarily of 1, an air cylinder, in which the air drawn from the atmosphere is compressed; 2, a steam cylinder, located above the air cylinder, the two being connected by a suitable center piece; 3, steam and air pistons mounted on a common piston rod; and 4, a valve motion controlling steam admission and exhaust.

The compressor is double acting, steam being admitted alternately on either side of the steam piston which, being directly connected with the air piston, causes both to move up and down.

On the upward stroke of the air piston, the air above it is compressed and discharged into the main reservoir, while the space below is filled with air drawn from the atmosphere. On the downward stroke, this operation is reversed. The steam exhaust is piped to the smoke stack or to the exhaust cavity of the saddle.

The steam valve should be located in the cab on the engineer's side of the steam turret with the handle in a convenient position for operation.

The steam cylinder lubricator connection is taken off from the pipe between the steam valve and governor.

A compressor governor is placed in the steam supply pipe between the lubricator and the compressor, its function being to start and stop the compressor automatically within predetermined pressure limits.

Compressor Troubles.—The following are the disorders most common in compressor operation, with causes and remedies:

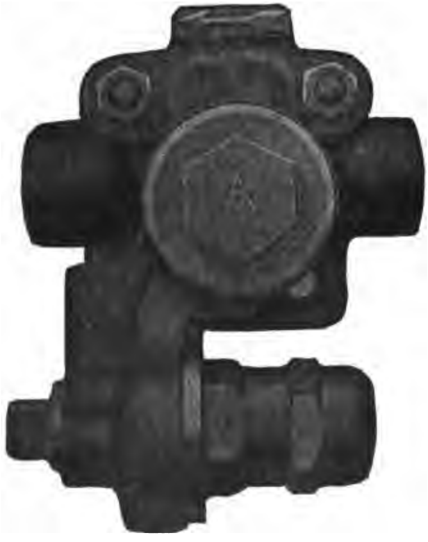


FIG. 2,014.—Westinghouse C–6 reducing valve. This valve is the well known feed valve that has been used for many years in connection with the G–6 brake valve, but in the ET equipment, it is attached to a pipe bracket. The only difference between it and the B–6 feed valve is in the adjustment, it being designed to reduce main reservoir pressure to a single fixed pressure, which in this equipment is, as already stated, 45 pounds. ***To adjust this valve,*** remove the cap nut on the end of the spring box; this will expose the adjusting nut, by which the adjustment is made. It is called a "reducing valve" when used with the independent brake and air signal systems, simply to distinguish it from the feed valve supplying the automatic brake valve.

Compressor refuses to start. CAUSE:—insufficient oil, from scant or no feed or working water; leaky piston rings in the small ends of the main valve piston; or rust having accumulated during time compressor has lain idle. REMEDY:—Shut off steam, take off reversing valve cap, pour in a small quantity of valve oil, replace cap, and then turn on steam quickly. In many cases when the compressor will not start when steam is first turned on, if steam be then turned off and allowed to remain off one or two minutes, and then turned on quickly, it will start without the use of any oil, except that from the lubricator.

Compressor groans. CAUSE:—1, air cylinder needs oil. REMEDY: 1, put some valve oil in air cylinder. CAUSE:—2, piston rod packing rod dry and binding. REMEDY:—2, saturate piston swab with valve oil. CAUSE:—3, steam cylinder needs oil. REMEDY:—3, increase lubricator feed.

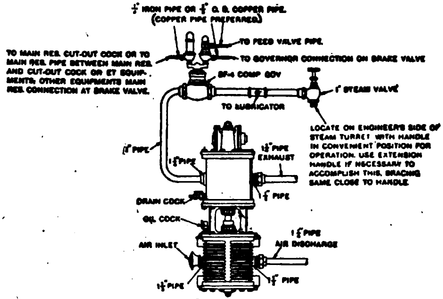
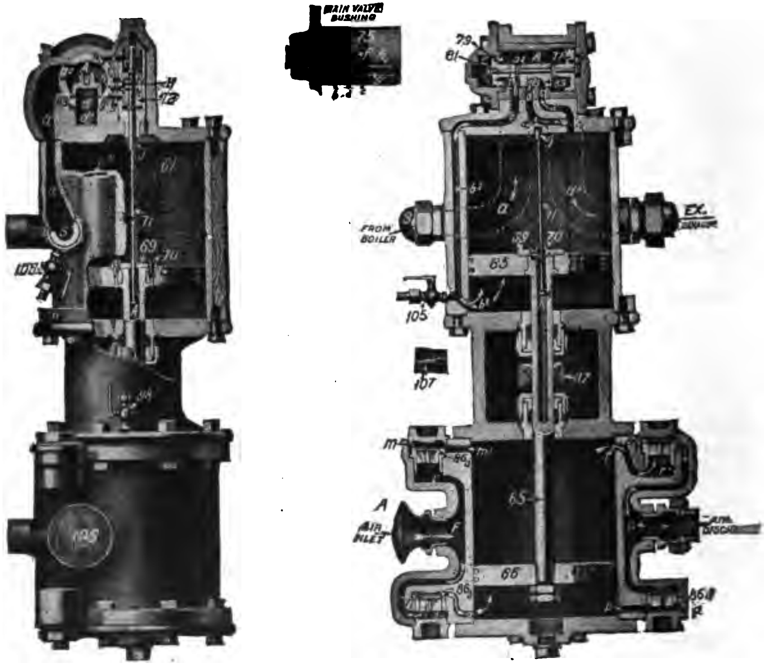

½" IRON PIPE OR ⅝" O. D. COPPER PIPE. (COPPER PIPE PREFERRED)
TO FEED VALVE PIPE.
TO MAIN RES. CUT-OUT COCK OR TO MAIN RES. PIPE BETWEEN MAIN RES. AND CUT-OUT COCK OR ET EQUIPMENTS; OTHER EQUIPMENTS MAIN RES. CONNECTION AT BRAKE VALVE.
TO GOVERNOR CONNECTION ON BRAKE VALVE
SF-4 COMP GOV
1" STEAM VALVE
TO LUBRICATOR
LOCATE ON ENGINEER'S SIDE OF STEAM TURRET WITH HANDLE IN CONVENIENT POSITION FOR OPERATION. USE EXTENSION HANDLE IF NECESSARY TO ACCOMPLISH THIS, BRACING SAME CLOSE TO HANDLE.
1" PIPE
1½" PIPE
1½" PIPE EXHAUST
1½" PIPE
DRAIN COCK
OIL COCK
1½" PIPE
AIR DISCHARGE
AIR INLET
1½" PIPE
1½" PIPE
9½" AIR COMPRESSOR

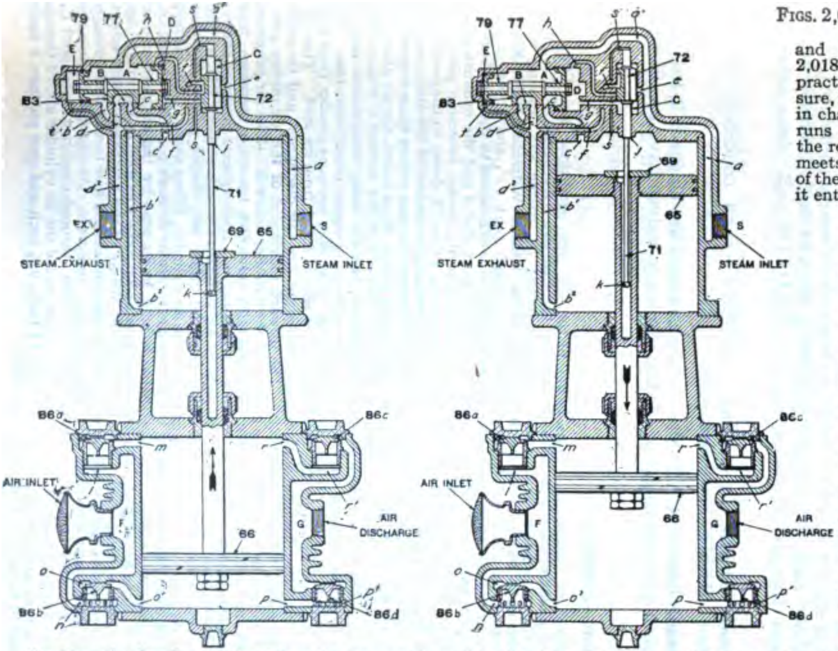
FIG. 2,015.—Installation diagram of Westinghouse 8 or 9½ inch steam driven air compressor. Piping. All pipes should be hammered to loosen the scale and dirt, have fins removed, and be thoroughly blown out with steam before erecting; bends should be used wherever possible instead of ells, and all sags avoided. Shellac or Japan varnish should be applied on the male threaded portion only, and never in the socket. Do not use red or white lead. The figure shows the recommended arrangement and sizes of the piping for one compressor. The size of pipe, particularly of the steam supply pipe, should never be smaller than that indicated in order to obtain maximum efficiency from the compressor. The governor should be located in the steam supply pipe between the lubricator connection and the compressor in order to insure its receiving the necessary lubrication. The lubricator connection consists of a tee, the side outlet of which connects to the lubricator. In two-compressor installations, the governor should be located in the main steam supply pipe between the lubricator fitting and the steam branch pipe leading to each compressor.

Excessive leakage past the air piston packing rings or past a discharge valve causes heating, destroys lubrication, and results in groaning.

Uneven strokes of the compressor. CAUSE:—probably: 1, excessive leakage past air piston packing rings and sticky air valves; 2, improper lift of air valves; 3, clogged discharge valve passages; 4, leaky air valves; or 5, binding or cutting of the reversing rod. REMEDY:—locate cause, if



Figs. 2,016 to 2,020.—Westinghouse air compressor; sectional and diagrammatic views. Fig. 2,016, section through reversing valve; fig. 2,017, main valve bushing; fig. 2,018, section through main valve; fig. 2,019, up stroke diagram; fig. 2,020, down stroke diagram; figs. 2,019 and 2,020, are distorted to show as clearly as possible the connections of the various ports and passages but not the actual construction of the parts. ***Steam end.*** Considering first the steam end of the compressor, steam from the supply enters at the connection marked "from boiler," or "steam inlet," and flows through the passageways *a*, a^2, to the chamber *A*, above the main valve 83 and between the pistons 77 and 79, and through passage *e*, to chamber *C*, in which is reversing valve 72. The supply and exhaust of steam to and from the steam cylinder is controlled by the main valve 83, which is a "D" type of slide valve. It is operated by the two pistons 77 and 79, of unequal diameters and connected by the stem 81. The movement of these two pistons and the main valve is controlled by the reversing valve 72, which is in turn operated by the main steam piston 65, by means of the reversing rod 71, and the reversing plate 69. As will be seen from the following description, the duty of the reversing valve 72, is to alternately admit steam to or discharge it from chamber *D*, at the right of piston 77, thus alternately balancing or unbalancing this piston. The reversing valve is operated by the reversing rod 71. This rod is alternately moved up and down by reversing plate 69, which engages reversing shoulder *j*, fig. 2,016, on the upward stroke of the steam piston and button *k*, at the end of the rod, on the downward stroke. Chambers *A* and *C*, are always in free communication with each other and with the steam inlet through port *e*, *e'*, as shown in the cuts. Live steam is therefore always present in these chambers, *A* and *C*. Chamber *E*, at the left of small piston 79 is always open to the exhaust passage *d*, through the ports *t* and *t'*, shown in the main valve bushing, fig. 2,016



FIGS. 2,016 to 2,020.—*Continued.*

and diagrammatically in figs. 2,018 and 2,019. Exhaust steam, practically at atmospheric pressure, is therefore always present in chamber *E*. A balancing port runs diagonally to the right in the reversing valve cap nut, and meets a groove down the outside of the reversing valve bush, where it enters the upper end of the cylinder through a small port in the head. The object of this is to assure the same pressure above as below the reversing rod, whether there be live or exhaust steam in the upper end of the cylinder, thus balancing it so far as steam pressure is concerned. When reversing slide valve 72, is in its lower position, as in figs. 2,016 and 2,019. Chamber D, is connected (through ports *h'*, *h*, reversing valve exhaust cavity H, fig. 2,016, and ports *f* and *f*,) with main exhaust passage *d*,*d'*,d^2, and there is, therefore, only atmospheric pressure at the right of piston 77. Therefore, as chamber E, at the left of piston 79, and chamber *D*, at the right of piston 77, are then both connected to the exhaust, as already explained, the pressure of the steam in chamber A, has driven the larger piston 77, to the right, and it has pulled the smaller piston 79, and the main valve 83, with it to the position shown in figs. 2,019 and 2,020. The main valve 83, is then admitting steam below piston 65, through port *b*,*b'*,b^2. Piston 65, is thereby forced upward, and the steam above piston 65, passes through port *c'*,*c*, exhaust cavity B, of main valve 83, port *d*, and passage *d'*,d^2, to connection *Ex*, at which point it leaves the compressor and discharges through the exhaust pipe into the atmosphere. When piston 65, reaches the upper end of its stroke, reversing plate 69, strikes shoulder *j*, on rod 71, forcing it and reversing slide 72, upward sufficiently to open port *g*, (figs. 2,016 and 2,020). Steam from chamber C, then enters chamber D, through port *g* and port *g'*, of the bushing. The pressures upon the two sides of piston 77, possible, and correct it by cleaning out clogged or dirty passages and air valves, replacing worn or leaky valves or rings or straightening or replacing the reversing rod.

Slow in compressing air. CAUSE:—1, leakage past the air piston packing rings, due to poor fit, or wear in cylinder or rings; 2, valves and passages dirty; or, 3, air suction strainer clogged. REMEDY: 1 and 2, to determine which is causing the trouble, obtain about 90 lbs. air pressure, reduce the speed to from 40 to 60 single strokes per minute, then listen at the "air inlet" and note if air be drawn in during only a portion of each stroke, and if any blow back. If the latter, an inlet valve is leaking. If the suction do not continue until each stroke is nearly completed, then there is leakage past the air piston packing rings or back from the main reservoir past the air discharge valves. REMEDY:—3, clean strainer thoroughly.

Compressor erratic in action. CAUSE:—worn condition of the valve motion. REMEDY:—replace it.

Compressor heats. CAUSE:—1, air passages are clogged; 2, leakage past air piston packing rings: or, 3, the discharge valves have insufficient

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FIGS. 2,016 to 2,020.—*Continued.*

are thus equalized or balanced. Considering piston 79, the conditions are different. Chamber E, as already stated, is always open to the exhaust. As piston 77, is now balanced, the steam pressure in chamber A, forces piston 79, to the left drawing with it piston 77, and main valve 83, to position shown in fig. 2,020. With main valve 83, in this position steam is admitted from chamber A, through port c,c', above piston 65, forcing it down; at the same time the steam below this piston is exhausted to the atmosphere through port b^2,b',b, exhaust cavity B, in the main valve, port d^2,d',d and the exhaust pipe connected at Ex. When piston 65, reaches the lower end of its stroke, reversing plate 69, engages reversing button k, and draws rod 71, and reversing valve 72, down to the positions shown in figs. 2.018 and 2,019, and one complete cycle (two single strokes) of the steam end of the compressor has been described. Air end. The movement of steam piston 65, is imparted to air piston 66, by means of the piston rod 65. As the air piston 66, is raised, the air above it is compressed, and air from the atmosphere is drawn in beneath it; the reverse is true in the downward stroke. On the upward stroke of piston 66, the air being compressed above it is prevented from discharging back into the atmosphere by upper inlet valve 86*a*. As soon as the pressure in ports r,r', below upper discharge valve 86*c*, becomes greater than the main reservoir pressure above it, the discharge valve 86*c*, is lifted from its seat. The air then flows past this valve down through chamber G, out at the "air discharge" and through the discharge pipe into the main reservoir. The upward movement of the air piston produces a partial suction or vacuum in the portion of the cylinder below it. The air pressure below piston 66, and on top of the lower left-hand inlet valve 86*b*, becomes, therefore, less than that of the atmosphere in port n, underneath this valve. Atmospheric pressure therefore raises valve 86*b*, from its seat, and atmospheric air is drawn through strainer 106, at the "air inlet," into chamber F, and port n, below the inlet valve 86*b*, thence past that valve and through ports o and o', into the lower end of the air cylinder, filling same. Air cannot enter this part of the cylinder by flowing back from the reservoir through passages D and G, and lower discharge valve 86*d*, since this valve is held to its seat by the main reservoir pressure above it. The lower inlet valve 86*b*, seats by its own weight as soon as the up stroke of the air piston 66, is completed. On the downward stroke of the compressor, the effect just described is reversed, the air below piston 66, being compressed and forced out through ports p and p', past lower discharge pipe into the main reservoir. At the same time air is being drawn in from the atmosphere through the "air inlet" through chamber F and port l', upper inlet valve 86*a*, and ports m and m', into the upper end of the air cylinder above the air piston 66. The inlet and discharge valves of the 11 inch, 9½ inch and new pattern 8 inch compressors should each have a lift of ⅛ of an inch.

lift. REMEDY:—1, clean air passages; 2, renew air piston rings; 3, regulate lift of discharge valves to 3/32 of an inch. A compressor in perfect condition will become excessively hot and is liable to be damaged if run very fast and continuously, for a long time.

Compressor Pounds. CAUSE:—1, air piston is loose; 2, compressor not well secured to boiler, or causes some adjacent pipe to vibrate; 3, the reversing plate, 69, is loose; or, 4, the reversing rod or plate may be so worn that the motion of the compressor is not reversed at the proper time. REMEDY:—repair and renew worn parts and tighten loose connections.

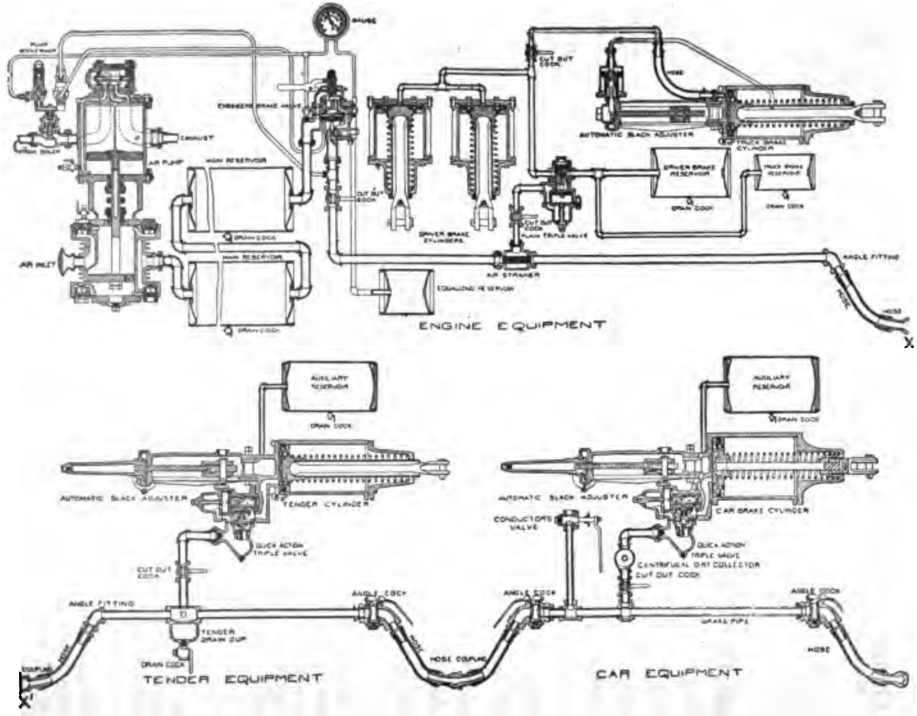
Points Relating to Compressors.—In connection with the problem of good maintenance for steam driven air compressors of this type, the heating of the air cylinder incident to air compression is perhaps the most important. The operation of the compressor continuously at high speeds or against excessive pressures inevitably results in high temperatures which tend to destroy the lubrication, causing the air cylinders to cut, and the groaning of the air compressor, besides filling the discharge passages with deposits from burnt oil, producing undesirable condensation of moisture in the brake system, and in general, reducing the over all efficiency of the compressor.

Under normal conditions, the speed should not exceed 140 strokes per minute and such a speed should not be maintained continuously for any considerable time, as even this speed will eventually cause excessive heating. Continuous running at high speed will cause excessive heating of the air end of the compressor. Overheating from this cause is an indication that a compressor of larger capacity is required.

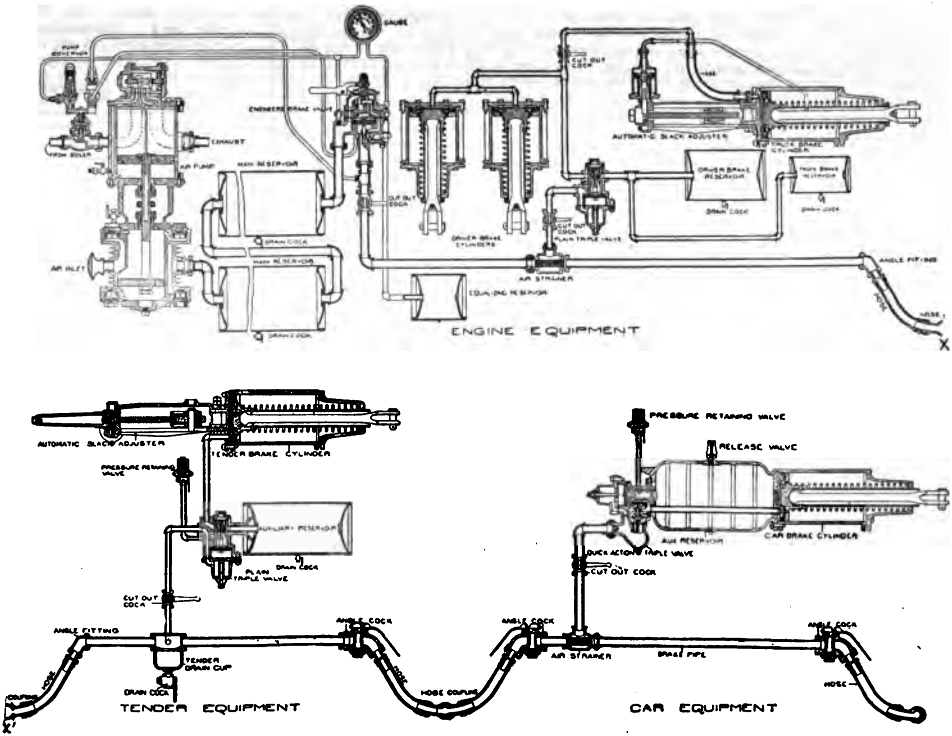
It is therefore desirable, first, that the compressor be of ample capacity for the service desired; second, that it be well lubricated and otherwise maintained in good condition; and third, that leakage from any source whether within the air compressor itself or in the brake system be minimized in every practical way.

One of the most serious leaks is through the air cylinder stuffing box as it not only greatly decreases the air delivered and, by the faster speed required, increases the heating, but it also causes pounding through loss of cushion. When tightening the packing, do not bind the rod, as to do so will damage both the packing and the rod. Be careful not to cross the gland nut threads.

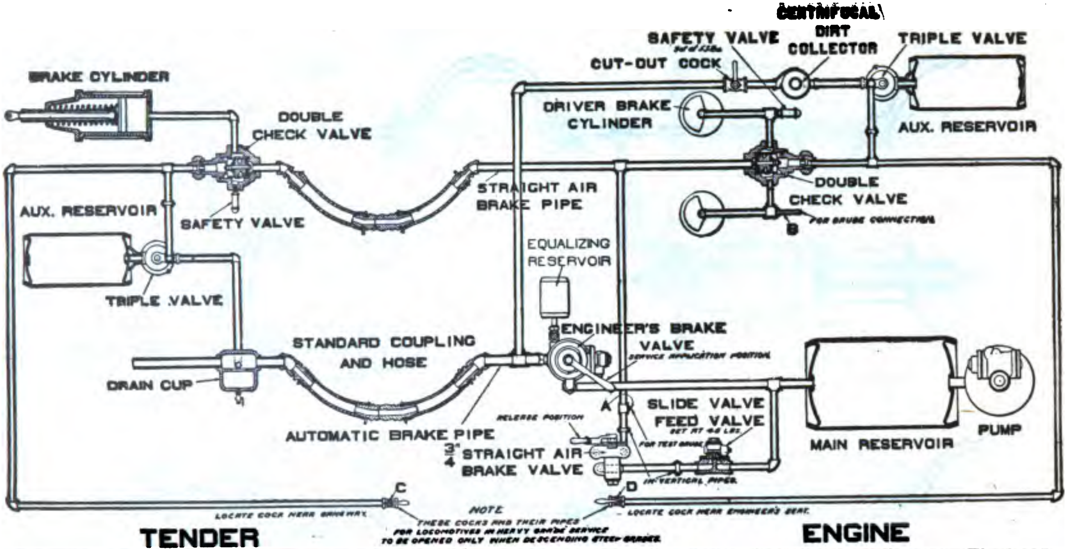
With two compressors per engine, the separate throttles should be kept



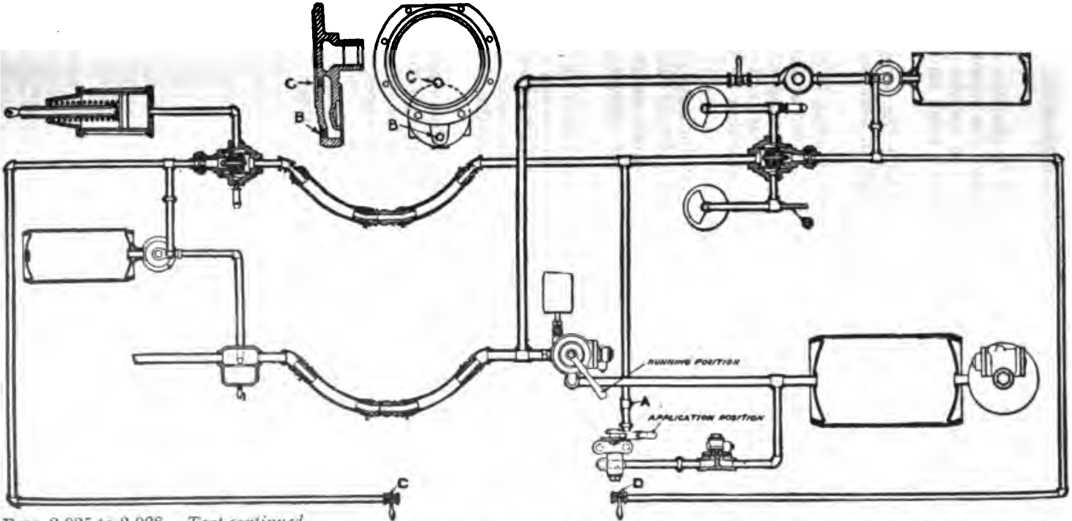
FIGS. 2,021 and 2,022—Westinghouse quick action automatic brake equipment for passenger service.



FIGS. 2,023 and 2,024.—Westinghouse quick action automatic brake equipment for freight service.



FIGS. 2,025 to 2,028.—General arrangement of Westinghouse combined automatic and straight air brake equipment. Fig. 2,025, freight locomotive equipment when automatic brake is applied; fig. 2,028, freight locomotive equipment when straight air brake is applied; figs. 2,026 and 2,027, pressure head. The straight air supply is taken from the main reservoir pipe between the main reservoir and the automatic brake valve so as to insure clean, dry air. Before reaching the straight air brake valve it must pass through the reducing valve, set at 45 lbs., this consisting of a standard slide valve feed valve, attached to a special pipe connection made for the purpose. This reducing valve should always be located in the cab and preferably at a point where in cold weather it will prevent freezing. A special cylinder pressure head shown in figs. 2,026 and 2,027 is now furnished for all tenders having quick action triple valves. For ordinary equipment the bosses A and B, are not drilled, and port C, the opening into the brake cylinder from the triple valve, is not tapped. In order to change from the ordinary to the combined automatic and straight air, it is necessary to have the port C, tapped for ¾ inch pipe plug, and the plug faced off flush on the cylinder side of the head. Either of the bosses A or B, may be drilled and tapped for ½ inch pipe, whichever be the more convenient. ***The double check valve,*** on engine and tender is connected as shown. In either applying or releasing the brakes with the straight air or automatic, the air must pass through this valve. The two side openings of the double check valve are brake cylinder connections, joined by a cored passage way. ***The safety valve*** on the



FIGS. 2,025 to 2,028.—*Text continued.*

tender may be screwed into one of the cylinder conections of the double check valve as indicated in the cuts, or in the pipe between the double check valve and the cylinder. On the engine one brake cylinder should be connected to each of the double check valve cylinder connections, and the safety valve in either driver brake cylinder pipe. Each safety valve should be adjusted to open at 53 lbs. It should be in direct communication with brake cylinder pressure whether the automatic or straight air be used, and should be placed vertically upward so that dirt and water cannot accumulate inside. As the straight air should never give over 45 lbs. cylinder pressure, and the automatic not over 50 lbs., a correctly adjusted safety valve will never operate unless an improper condition exists, but under the latter will guard against a dangerously high cylinder pressure. The cocks C and D, with their pipes are only for locomotives operating on heavy mountain grades. The cocks are furnished only when specially ordered. They are to be opened only on descending very long grades and should be closed at once on reaching the end of the grade. Cock C, should be located adjacent to the gangway so it can be operated when running, and D, near the engineer's seat. In descending heavy grades both are left open. The driver and tender auxiliaries are recharged with those of the train, but automatic application is prevented on the brakes of the former, thus permitting of the greatest use practicable, without danger of loosening tires of the tender and driver brakes when recharging the train brakes, the critical operation in braking down steep grades. A gauge for the straight air is necessary for satisfactory service. It should be connected so as to show brake cylinder pressure in automatic as well as straight air application.

wide open and the speed regulated by the main compressor throttle. The purpose is to divide the work equally.

If necessary to replace a broken air valve on the road or elsewhere not permitting of proper fitting, at the earliest opportunity have the repairman replace the temporary valve with another so as to insure the correct angle and width of valve and seat contact, the needed ground joint and the requisite lift.

Never remove or replace the upper steam cylinder head with the reversing valve rod in place as to do so will almost invariably result in bending the rod. A bent rod is very liable to cause a "pump failure."

It is evident that a compressor cannot compress more air than it draws in and not that much if there be any leakage to the atmosphere about the air cylinder. Bearing this in mind, practice frequently listening at the "air inlet" when the compressor is working slowly while being controlled by the governor, and wherever a poor suction is noted on either or both strokes locate and report the fault.

Any unusual click or pound should be reported as it may indicate either a loose piston or a reversing valve plate cap screw or other serious fault.

Any steam leakage that can reach the air inlet of the compressor should be promptly repaired as such increases the danger of water entering the brake pipe.

Keeping the suction strainer clean is of the utmost importance, as even a slightly clogged strainer will greatly reduce the capacity where the speed is at all fast. A seriously or completely obstructed strainer, as by accumulated frost, aggravated by rising steam, will increase the compressor speed and will also be indicated by inability to raise or maintain the desired pressure.

It is an aid to good operation to thoroughly clean the air cylinder and its passages at least three or four times a year, by circulating through them a hot solution of lye or potash. This should always be followed by sufficient clean, hot water, to thoroughly rinse out the cylinder and passages, after which a liberal supply of valve oil should be given the cylinder. Suitable tanks and connections for performing this operation can easily be arranged in portable form. Never put kerosene oil in the air cylinder to clean it.

CHAPTER 38

MARINE ENGINES

There are many types of marine engine, and these are due, not only to the varied conditions of service, but to the gradual increase in steam pressures resulting from improved methods and better materials used in boiler construction. The numerous types of marine engine may be classed:

1. With respect to the initial pressure, as

a. Low pressure { **walking beam paddle engines.**

b. Medium pressure { **Simple and compound.**

c. High pressure { **triple and quadruple expansion engines.**

NOTE.—***Robert Fulton,*** though not the first to propel a vessel by steam, was the first to make steam navigation a success. Unlike those who abandoned their work after one or two unsuccessful trials he had perseverance enough to continue until his efforts were successful. In 1803, while in Paris, he constructed a small steam boat the trial trip being made on the Seine. The experiment was so successful that he had an engine built in England and then returned to America. In the spring of 1807 the ***"Clermont"*** was launched and the English engine was put aboard in August. This boat was 133 feet long, 18 feet beam and 7 feet in depth. The "Clermont" made a trip from New York to Albany (about 145 miles) in 32 hours and returned in 30 hours. The sails were not used on either occasion. This was the first successful long trip ever made by a steam boat. The engine of the "Clermont" was coupled to the crank shaft by a bell crank and the paddle wheels were connected to the crank shaft by gearing. The cylinders were 24×48. Fulton afterward built several steamers the largest one measuring 2,475 tons. This vessel was built for the United States Navy and was a very large steamer for that period.

NOTE.—While Fulton was building steamers with success, ***Stevens*** of Hoboken built a steamboat which showed great merit. He used a horizontal sectional water tube boiler with a working pressure of over 50 lbs. per sq. in., the usual pressure at that time being 5 to 7 lbs. His engine was direct acting, condensing. The most remarkable feature was the use of a screw propeller of four blades.

2. With respect to the stages of expansion, as
    - *a.* Simple;
    - *b.* Compound
        - steeple;
        - fore and aft;
        - three cylinder.
    - *c.* Triple expansion
        - three cylinder
            - single
            - steeple;
        - four cylinder
            - double
            - steeple;
        - five cylinder
            - double
            - steeple;
        - six cylinder
            - triple
            - steeple;
    - *d.* Quadruple expansion
        - four cylinder;
        - five cylinder;
        - six cylinder
            - triple
            - steeple;
3. With respect to the method of operating, as
    - *a.* Non-condensing;
    - *b.* Condensing.
4. With respect to the application of the power, as
    - *a.* Screw or propeller;
    - *b.* Paddle
        - walking beam;
        - stern wheel.
5. With respect to the position of the cylinder or cylinders, as
    - *a.* Horizontal;
    - *b.* Vertical
        - back acting;
        - inverted;
    - *c.* Inclined.
6. With respect to the moving parts as
    - *a.* Reciprocating
        - single acting;
        - double acting,

NOTE.—In ***Stevens' second boat,*** the engines were of the same type as before, but twin screws were used instead of a single screw.

NOTE.—Although the screw propeller was introduced as early as 1804, it was practically given up and for about thirty years the paddle wheel was the principal means of propulsion.

b. Oscillating { single acting; double acting,
c. Rotary;
d. Turbine.

7. With respect to the division of power, as

a. Single;
b. Double { duplex; twin;

8. With respect to the cylinders, as

a. Lagged;
b. Jacketted.

9. With respect to the frame, as

a. Closed;
b. Semi-open;
c. Open.

10. With respect to duty, as

a. Light duty { high speed;
b. Medium duty { moderate speed.
c. Heavy duty { slow speed.

etc., etc.

Characteristics of Marine Loads.—Marine engines operate most of the time at full load, the full load rating being higher than in the case of stationary engines, that is, the mean effective pressure and piston speed are higher in order to reduce the weight and space occupied by the engine. Thus more room is available for passengers and freight.

The marine engine then, is inherently a heavy duty machine as compared with stationary engines, and because of the high piston speeds and high mean effective pressures employed direct non-releasing valve gear are used which cut off comparatively late, the standard cut off being six-tenths stroke in multi-stage expansion engines. With this late cut off, the expansion is not continuous but there is considerable drop in each cylinder, and also because of this late cut off the cylinder ratios are greater than in stationary engines for a given number of expansion. The heavy duty calls for a rugged construction throughout, the bearings being large, connecting rods short with large ends, etc.

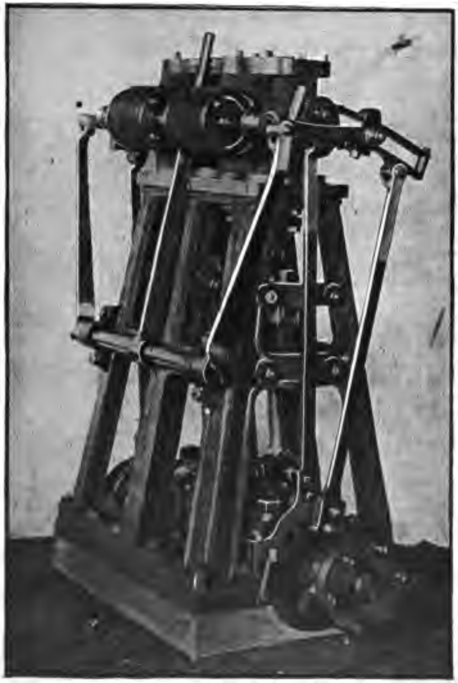


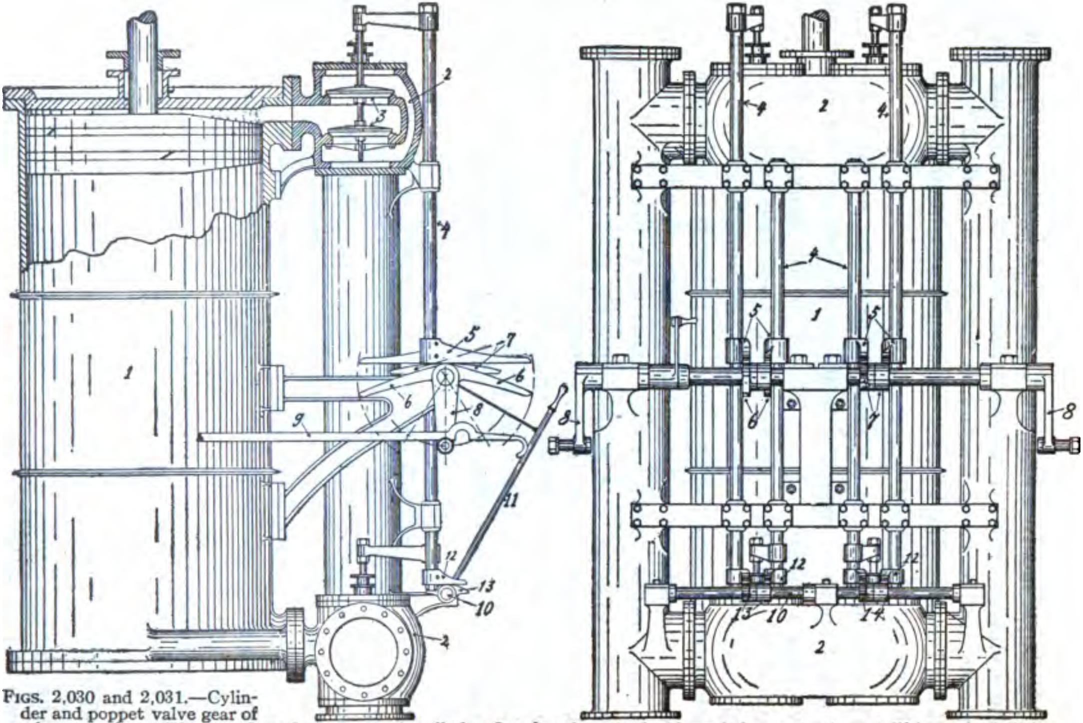
Fig. 2,029.—Rees two cylinder (double) propeller engine, with rocking valves, link motion and cast iron frames.

Walking Beam Engines.—This type of engine occupies considerable space for its power because it is operated at low pressure and low number of revolution. It is well known on the lakes, sounds, bays and rivers of the United States, has usually only a single cylinder, with a stroke quite long in proportion to its diameter, and varying frequently between 7 and 12 feet. The walking beam engine is a good example of condensing operation with low initial pressure.

The pressures in use are from 25 to 75 lbs. at the boiler; and the usual speed is from 20 to 30 revolutions per minute. The valve gear is almost always of the double poppet valve type, actuated by cams on a rock shaft. To secure expansion in the single cylinder, some form of cut off arrangement is provided, of which constructions the Sickels cut off gear is a well known one. Under the influence of a second eccentric, it allows the poppet valve to drop precisely by disconnecting it from the lifting rod. Reversal is effected by disconnecting the eccentric rods from the rock shaft lever and giving live steam by hand on the opposite side of the piston. This entire suspension of the automatic action of the valve, which would be impossible in quick running engines, proves exceedingly handy and serviceable in the beam engine.

Where long continued backing is required, as in ferry boats, special backing eccentrics and rods are provided, which hook in when the forward eccentric rods hook out. Some engines have the loose eccentric reversing gear which consists of a single eccentric, which is free to turn upon the shaft and a disc keyed to the shaft, which is provided with a circular groove in which slides a block, attached to the eccentric. The ends of this groove limit the position of the eccentric to the proper angle it has to make with the crank. To reverse the engine, the valve gear has to be operated by hand for at least one half a revolution, for which a hand lever is used.

The general construction of the cylinder and valve gear is shown in figs. 2,030 and 2,031.



FIGS. 2,030 and 2,031.—Cylinder and poppet valve gear of a beam engine. ***The numbered parts are:*** 1, cylinder; 2, valve chest; 3, double seated poppet valves; 4, lifting rods; 5, lifting toes; 6, wipers for steam valves; 7, wipers for exhaust valve; 8, rock shaft arm; 9, eccentric rod; 10, small rock shaft; 11, lever to operate valve gear by hand; 12, lifting toes; 13, steam valve wipers; 14. exhaust valve wipers.

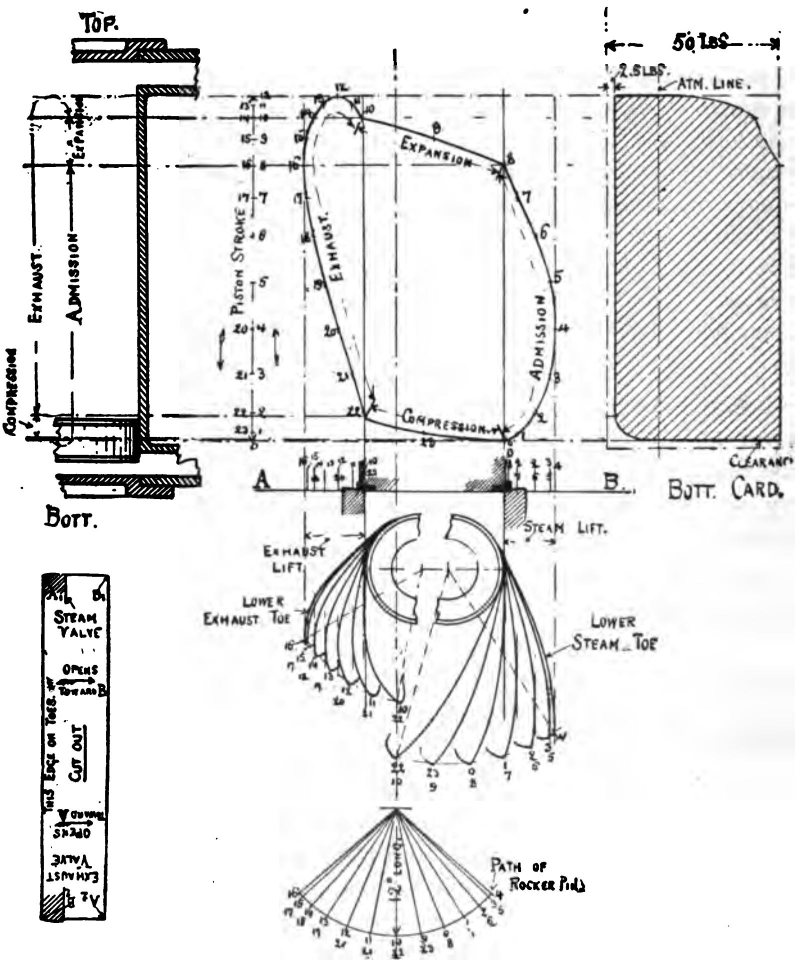
The valves 3, are balanced and lift squarely off these seats, thus, without friction. The rock shafts are operated by the eccentrics by means of the rock shaft arms 8, the one seen at the left in fig. 2,030, is for the steam valves, the other for the exhaust. To the rock shafts are attached the wipers 6, for the steam valves and 7, for the exhaust valves, which, when vibrated by the eccentrics, strike the lifting toes 5, which are attached to the lifting rods 4, raising these quickly, and thus by means of the attachment to the valve stems shown, raise the valves off their seats, thus admitting steam to the cylinder, and exhausting.

To operate the valve gear by hand, the hand lever 11, is released from the hook which supports it at rest in its midway position, and then vibrated it up and down. The steam valves are operated by means of the wipers 13, and the exhaust valves by means of wipers 14, attached to the small rock shaft 10, and striking the lifting toes 12.

After the engine is well started the eccentric rods 9, are released by means of a lever, shown on the rock shaft, which holds them clear of the rock shaft arms, and the hand lever is hooked up in its stationary position, remaining at rest while the engine is running. The engine shown is provided with loose eccentric reversing gear.

The valve chests 2,2, are bolted to the cylinder 1. In the position of the valves as shown, steam is passing through the column at the left (fig. 2,031,) and the exhaust through the right hand column. These columns also support the bearings for the rock shafts, their inner ends being supported by a bracket bearing attached to the cylinder.

The support of the engine is derived from the so called "A" frame that provides bearings for the walking beam pins at the top and for the shaft at one side, and connects to the cylinder



FIGS. 2,032 and 2,033.—Valve diagram for a beam engine with poppet valves. This diagram shows the relative positions of the piston and poppet valve. The piston travels vertically, while the toes swing horizontally, lifting the valves in the same direction. The valve rests, in the diagram, directly on the toes. The cut of slide valve, at the left of the diagram, is to be placed in vertical position and moved horizontally, resting successively upon the toe in the positions from 1, around to 7, for the steam, and on the other edge and other toe, from 10, around to 22, for the exhaust.

and guides on the other side, as shown in fig. 2,034. Great size and weight concentrated on a short length and a high center of gravity are characteristics in the beam engine that may finally cause it to be superseded by other constructions, regardless of the great popularity that it has enjoyed, except for vessels operating in shallow waters.

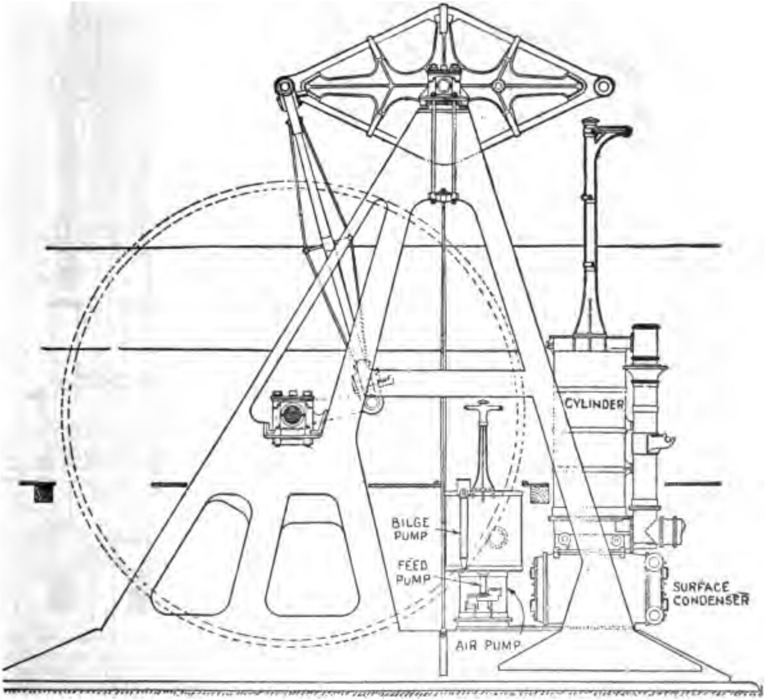
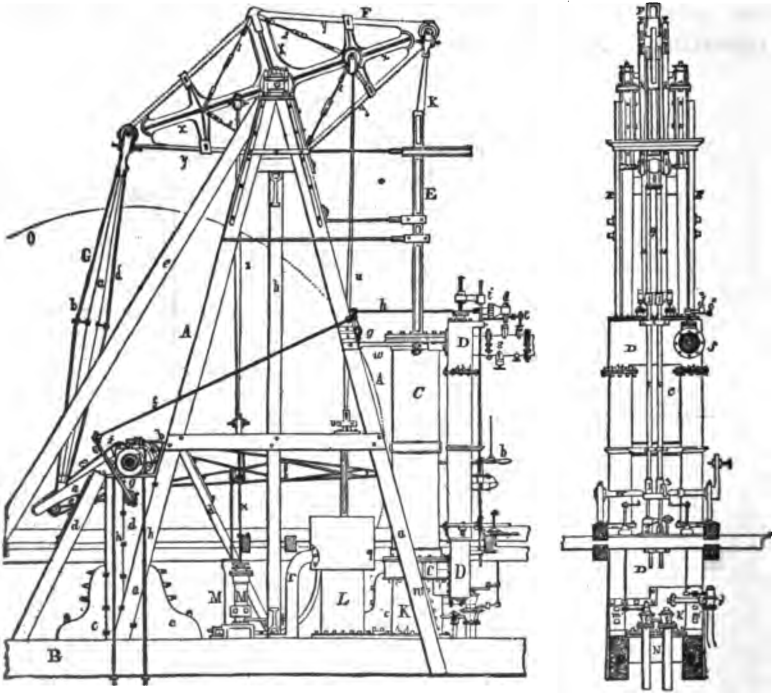


FIG. 2,034.—Diagram of walking beam engine showing arrangement of parts as cylinder, walking beam, A frame, condenser, pumps, etc.

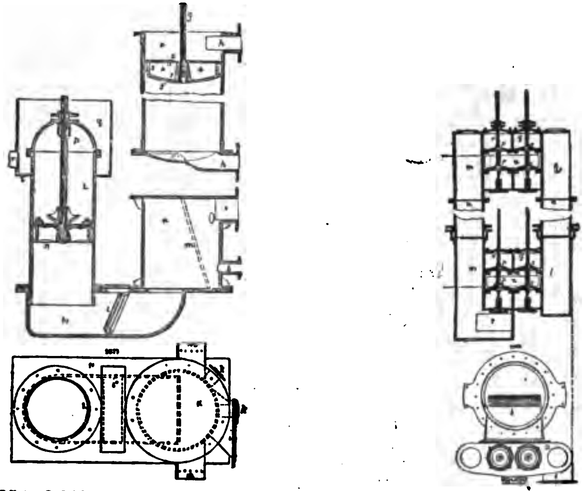
How to Set the Valves of a Walking Beam Engine.—Referring to figs. 2,030 and 2,031 and assuming that the rock shaft arm 8, is keyed on to the rock shaft in its proper relation to the center line of motion of the eccentric rod 9, and that the wipers 6, 7, are keyed on to the rock shaft, in their proper relation to the rock shaft arm, which is always the case in properly constructed engines, the first step is to ascertain the proper length of the eccentric rod, and the most convenient starting point for doing so is from the center of motion of the valve gear.

Hence, the first thing to be done in setting beam engine valves is to set and hold the rock shaft at the center of its motion, which is when the lifting



FIGS. 2,035 to 2,040.—Walking beam engine. Fig. 2,035, side elevation; fig. 2,036, end view; fig. 2,037, steam cylinder; fig. 2,038, air pump and condenser; fig. 2,039, plan of bed plate, showing the passage connecting the condenser with the air pump, and the opening by which the foot valve is introduced to its place; fig. 2,040, traverse section and plan of steam chests, showing arrangement of the balance valves. ***In construction,*** A, is the principal frame, B, keelsons; *aa*, frame legs; *b*, upright post; *c*, oak knees; *d*, supporting timbers; *e*, back stay; C, steam cylinder; C′ cylinder bottom; *f*, piston (under side); 1, is a solid web, rounded and in one piece with the center 2, by which it is keyed to the rod, and with the circular flange 3, at the circumference, upon which the packing is laid. These three parts are connected together by stiffening flanges, 4; and the whole is covered in by a flat plate 5, which holds down the packing, and is bolted to the body of the piston; *g*, piston rod; *hh*, steam ports; K, links; D, steam chests; *ll*, chambers whence the steam is admitted to the cylinder; *mm*, the chambers into which the rods, 4, are down, the valves, 3, seated, and the lifting toes, 5, adjusted the right distance from the rock shaft, and straight with each other, so that the ends of both wipers will be the same distance from their respective toes.

Next put the main crank on the center, and turn the throw of the eccentric directly in line toward the center of the eccentric hook pin; then make a fine center punch mark on the edge of toe pin, and one on the edge of the



Figs. 2,035 to 2,040.—*Text continued.* steam is discharged from the cylinder; *nn*, pipes connecting the upper and under chambers, bolted fast to upper chambers, but connected to the under chambers by expansion joints; *o*, steam valves; *p*, exhaust valves; *qr*, valve stems, having their lower ends guided in inverted caps, introduced through the under sides of the steam chests; their upper ends pass through stuffing boxes, and are connected on the outside to the brackets on the lifting rods; *s*, steam pipe; *c''*, throttle valve; *d''*, cut off valve, worked by a cam fixed on the crank shaft, which works on the lever *e''*, the fulcrum of which is fixed on the timbers of crank shaft bearings, working the lever on the valve spindle by means of the rod *f''*, the traverse shaft and levers *g''*, and the rod *h''*; *t*, exhaust passage, connected to the passage *i'*, in the condenser; *uu*, the steam passages to the cylinder; *vv*, the lifting rods, with brackets, 1 1, 2 2, fixed on them, and connected at the extremities to the valve spindles, on which they are adjusted by nuts; 3 3, the lifting faces; *w*, the traverse shaft; E, guides; F, walking beam; *y* and *z*, tension rods; *a''*, eccentric; *b''*, eccentric rod, having its length divided, at a point near the handle, into two parts jointed together, the longer and heavier part next the eccentric being supported on a vibrating joint. By this means it becomes an easy matter to disengage the rod from the lever of the traverse shaft, which is done by means of a small rope attached to the extremity of the rod. K, is the condenser; *i'*, exhaust steam passage; *k'*, injection passages. *l'*, injection cocks. L, air pump; *o*, bucket rod; *p'*, delivery valve. ***In operation***, when the bucket arrives at a certain height, the lid is raised and the water flows out all around, thus discharging more effectually and rapidly than by the common valve, and requiring little or no power to discharge, having only the lids to raise; *q'*, hot well, made of copper, and riveted to the air pump by means of a vertical flange; *r'*, waste pipe; *s'*, feed pipe; *t'*, hot well overflow pipe; *u'*, air pump drive rod; *v'*, air pump crosshead guide; M, feed pump; *x'*, pump rod, moving in the guides *y'y'*, driven from the beam by the rod *z'*. M', is the bilge pump; N, the bed plate; *i''*, the foot valve.

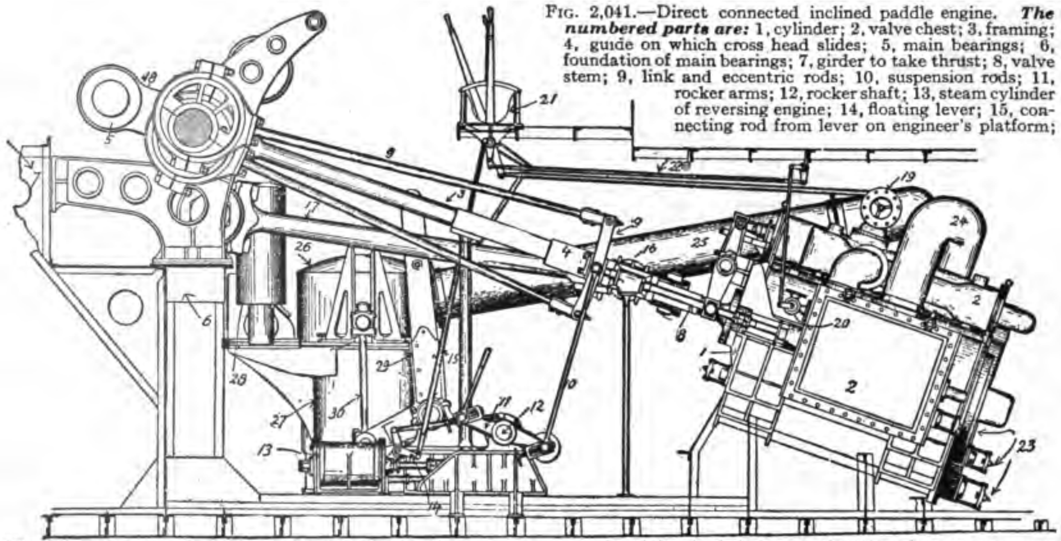


FIG. 2,041.—Direct connected inclined paddle engine. ***The numbered parts are:*** 1, cylinder; 2, valve chest; 3, framing; 4, guide on which cross head slides; 5, main bearings; 6, foundation of main bearings; 7, girder to take thrust; 8, valve stem; 9, link and eccentric rods; 10, suspension rods; 11, rocker arms; 12, rocker shaft; 13, steam cylinder of reversing engine; 14, floating lever; 15, connecting rod from lever on engineer's platform; 16, main engine cross head; 17, main engine connecting rod; 18, crank of paddle wheel shaft; 19, throttle valve; 20, pass over lever; 21 engineer's stand, with hand levers; 22, connecting rods from hand levers; 23, relief valves; 24, receiver piping; 25 exhaust pipe; 26, jet condenser; 27 air pump; 28, hot well; 29, air pump beams, bell crank shaft; 30, connecting rods to air pump cross head.

hook strap, and set a pair of compasses corresponding to the distance between those marks, and measure it; add to that distance half the throw of the eccentric; reset the compasses to that length, and move the eccentric until the center punch marks and compasses again correspond, and adjust the length of eccentric rod so that the hook will just engage the eccentric hook pin while the eccentric is held at that position; then slack up the rock shaft so that it can be moved, hook on the eccentric rod, and turn the eccentric in the direction to raise the required valve until it has the proper lead.

NOTE.—There can be no general rule given as to how the eccentrics should be placed in relation to the crank, as that depends on the relative arrangement of the lifters, valves, wipers, and rock shaft arms.

Now (if the engine be of the style that has two eccentrics and two rock shafts), proceed in the same manner with the exhaust valve gear and the valves are set.

To prove the accuracy of adjustment, turn the main crank to its opposite center, and if there be a difference in the lead, either lengthen or shorten the eccentric rod to make up half the difference, and turn the eccentric to make the other half, fasten the eccentric on the shaft and the valves will be right.

Inclined or Diagonal Paddle Engines.—This type is simply a horizontal engine set at such angle as to suit the height of the shaft at one end and the frames of the vessel at the other.

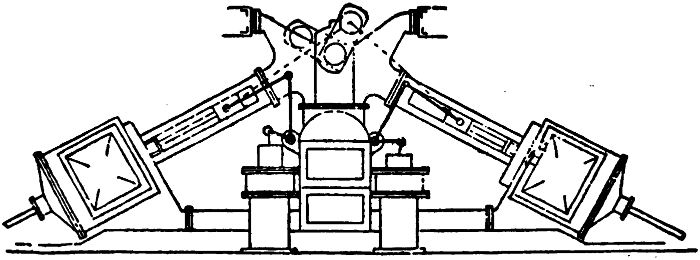


FIG. 2,042.—Double opposed diagonal engine showing arrangement of condenser and air pumps.

It takes up considerable distance in a fore and aft direction, and is somewhat heavier and more expensive than the other forms of paddle engine.

This engine is extensively used in Europe for paddle wheel propulsion, and conveniently allows compound and triple expansion arrangements, thereby ensuring economical use of high pressure steam.

The revolutions are usually higher than with the beam engine, while the stroke is correspondingly reduced. The valves and valve gears are similar to those of vertical screw engines, with link and radial gears in favor for reversing. The position in the hold of the ship allows a strong arrangement, with fairly good access to all parts.

Stern Wheel Engines.—There are many important runs to be made on shallow rivers and lakes, where owing to the lightness of draft required, no type of steamer can be satisfactorily used

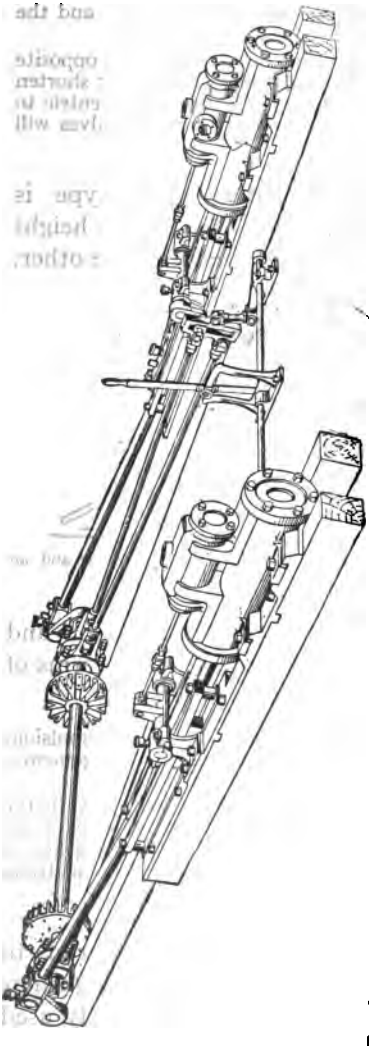


FIG. 2,043.—Direct connected stern wheel engine, showing general arrangement of the cylinders and driving gear, connected to the shaft of the stern paddle wheel.

except the stern paddle wheel. For many purposes, boats of this kind are altogether the most desirable that can be used.

The machinery consists of a boiler whose type is governed principally by the kind of fuel that is to be used.

The engines, two in number, are connected by wooden or metal connecting rods to the double-throw cranks arranged at ninety degrees on the ends of the paddle wheel shaft. They are reversed and controlled by a single lever, which may be located midway between the two engines or at one side of the boat, or may be located forward on the pilot house as desired.

The noticeable advancement that has taken place in the design and efficiency of small and medium size stern paddle wheel machinery is clearly due to the necessity for higher power and reduced weights, resulting in the extension of traffic on streams so shallow and rapid, that were formerly abandoned as impossible to navigate.

The engines usually have a stroke four times the cylinder diameter, while the valve gear used is of various types as shown in the accompanying cuts, slide, piston, and poppet valves being used.

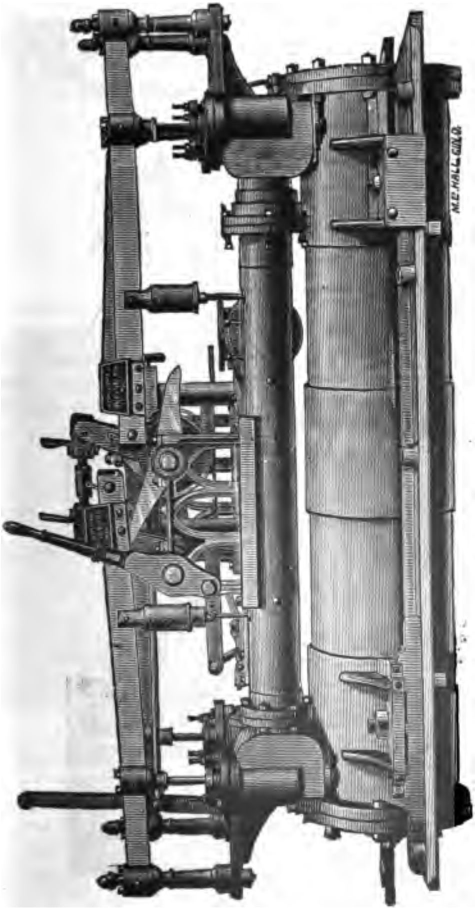


FIG. 2,044.—Reese high pressure stern wheel poppet valve lever engine.

Unbalanced slide valves are not desirable, especially for high pressures, because it was found that owing to the excessive load on the valve under high boiler pressure, the long eccentric rods would vibrate and back lash, thus having an unfavorable effect upon the steam distribution.

Poppet valves actuated by long cam driven levers are extensively used. The stroke of the engine as mentioned is long in proportion to the cylinder diameter, the standard ratio being 4 to 1. The usual range of sizes for high pressure engines range from 3¾ × 14, to 14 × 60.

The elements which enter into the design and construction of all successful steam craft are more numerous and conflicting when the boat is to be used for general business purposes on shallow, rapid rivers, than is commonly understood.

The machinery, equipment and power must be proportioned, not only to the hull, the load, and the runs to be made, but more especially to the shallow water and swift current, requiring careful distribution of the weights, while the character of the feed water and fuel must also be considered.

As to the best proportion of hull length, depth, and bearing lines, that depends on operating condition, draft of water, amount and consequent weight of power required.

The following table, which represents the practice of the Marine Iron Works of Chicago, will serve as a general guide in proportioning hull and engines. While the drafts of the several sizes is given very light it is recommended when conditions permit, to employ a deeper draft.

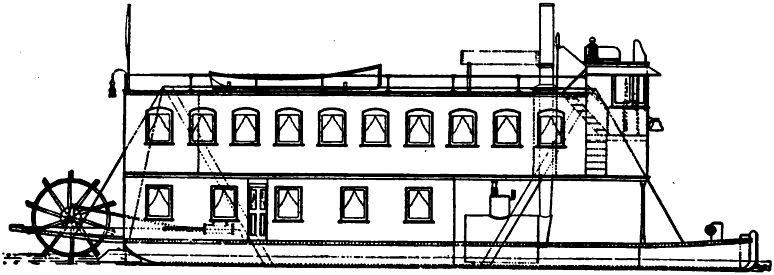


Fig. 2,045.—Type of Western river boat propelled by stern paddle wheel and direct, connected engine, such as is shown in the accompanying cut.

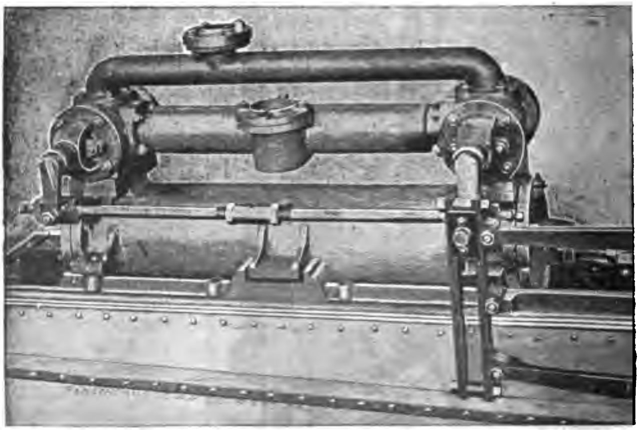
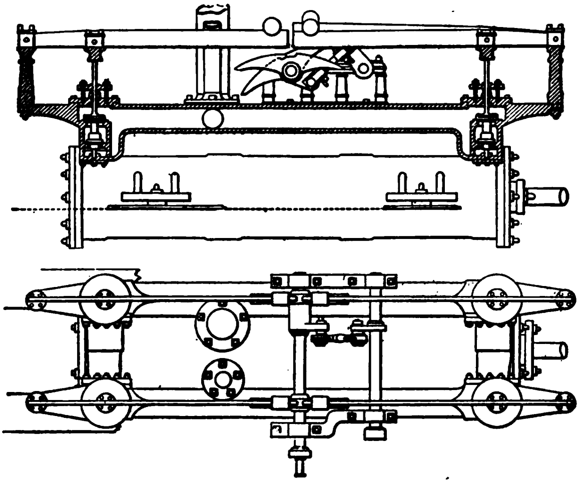


Fig. 2,046.—Rees high pressure stern wheel rotary valve engine.



FIGS. 2,047 and 2,048.—Sectional views of Western river engine with cam driven poppet valves.

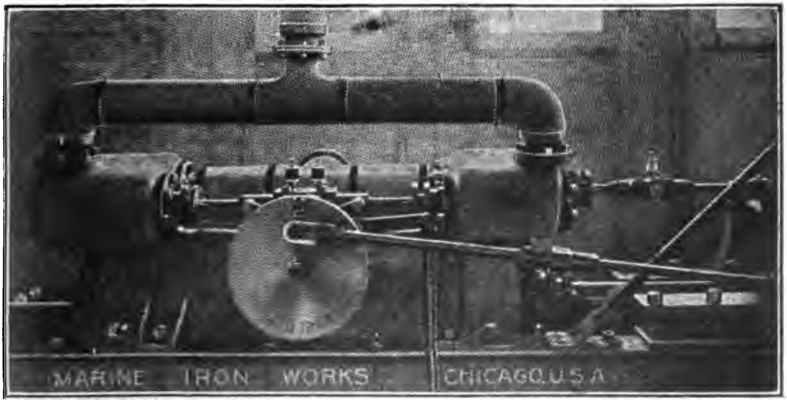
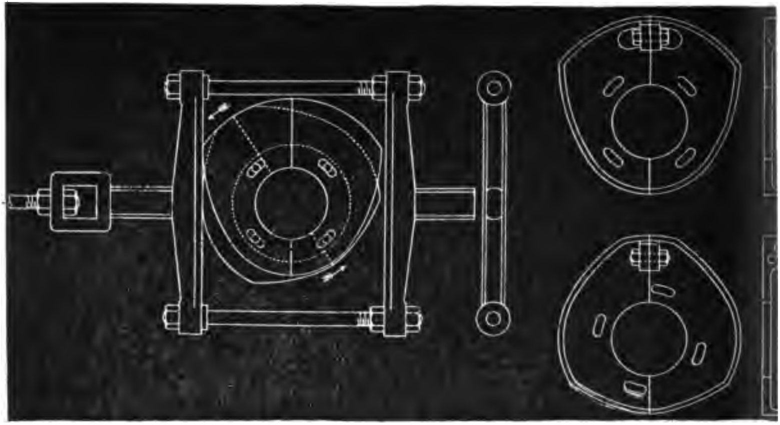


FIG. 2,049.—Marine Iron Works variable cut off valve. *Principal:* The cut off is changed by the method of *combined variable lap and variable throw.* ***In construction,*** the main valve chest and cut off valve chest (preferably in one casting) is so arranged as to allow a free and direct passage of steam to the piston. The lap of the cut off valve is adjusted by means of levers connected to valve rods, and to a sliding block in the disc. The valve rods being in two pieces are separated or spread as the block is moved upward, thereby adding to the lap and changing the travel at the same time. The disc is pivotly supported on a bracket centrally located between the steam chests, if there be two chests, or in the center if there be only one. The driving rod is connected to the main eccentric rods. The gear operates through its range of cut off without cramping the exhaust.



FIGS. 2,050 to 2,055.—Cam drive of Western river poppet valve engine. With this drive the motion of the valve may be made intermittent, giving a quick opening and closure, with intermediate periods of rest or very slow motion. In the ordinary type of gear two cams are used, known as the *full stroke*, and the *cut off* cam. When the engine is in full gear, the full stroke cam operates all four valves of the cylinder. The cut off cam is so arranged as to be hooked on after the full stroke cam has given headway to the boat, and is used in the go ahead motion only. This cam is so designed that steam may be cut off at any point as ¾, —, ½, etc. In the figure the form of full stroke is shown in full lines and the cut off cam in partly dotted lines. With this arrangement of gear the exhaust is opened and closed just at the end of the stroke, and hence neither early exhaust opening nor closure for cushion can be obtained. A means of obtaining the former has been found by blocking up the exhaust lifters somewhat, so that the valve will be slightly open when the engine is on the dead point. This insures an earlier opening of the exhaust and so clears the cylinder for the return stroke, but it gives likewise a later exhaust closure, so that with the engine on the center both exhaust valves are slightly open, and in full gear operation a slight "blow through" will occur. This disappears, however, when the cut off cam is engaged, because the opening movement of the latter is much slower than that of the full stroke cam. Various modifications of this simple cam gear have been introduced with a view of improving the general operation, especially by the provision of means for obtaining both steam and exhaust lead and compression, as well as independent movements for the go ahead and backing motions.

NOTE.—***In the Sweeney valve gear*** two full stroke cams are used, one for go ahead and one for backing, each set so as to give suitable exhaust lead and compression, while a separate cut off cam is fitted for the go ahead motion. The cross heads of these engines are usually of the locomotive type, with long brass gibs bearing on the top and bottom guides. The connecting rods are commonly of wrought iron or wood, with iron or steel fittings, and form one of the most peculiar features of these engines. Wood is often thus preferrred over metal because it seems to be better capable of standing the shocks and peculiar twisting strains which come upon the rod, and in spite of the strangeness of the combination, in some modern boats is found a nickel steel paddle shaft with a wooden connecting rod. The rods are very long, frequently as much as eight time the crank, and the best rods are made of Oregon fir, reinforced with brass straps which are let into the body of the rod and through bolted. The ends of the rods are fitted with brass boxes with straps, gibs, keys, etc.

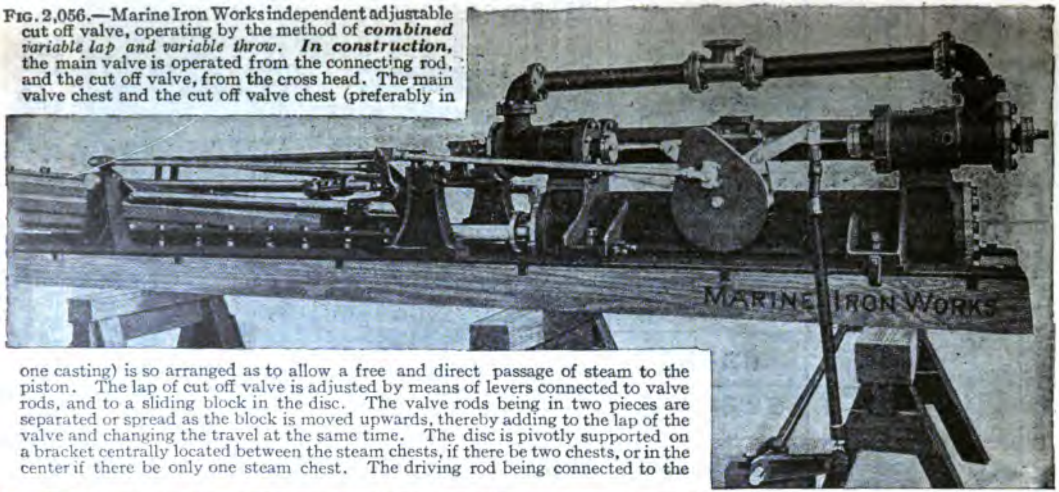


FIG. 2,056.—Marine Iron Works independent adjustable cut off valve, operating by the method of ***combined variable lap and variable throw.*** ***In construction,*** the main valve is operated from the connecting rod, and the cut off valve, from the cross head. The main valve chest and the cut off valve chest (preferably in one casting) is so arranged as to allow a free and direct passage of steam to the piston. The lap of cut off valve is adjusted by means of levers connected to valve rods, and to a sliding block in the disc. The valve rods being in two pieces are separated or spread as the block is moved upwards, thereby adding to the lap of the valve and changing the travel at the same time. The disc is pivotly supported on a bracket centrally located between the steam chests, if there be two chests, or in the center if there be only one steam chest. The driving rod being connected to the main eccentric rods. The hand lever which adjusts this cut off valve may be placed in any position convenient to the engineer. With this combination (reversing link and also cut off valve) neither the steam nor the exhaust ports are contracted, and the engine is relieved of back pressure. A full volume of steam is admitted to the cylinder without undue throttling or wire drawing of the steam.

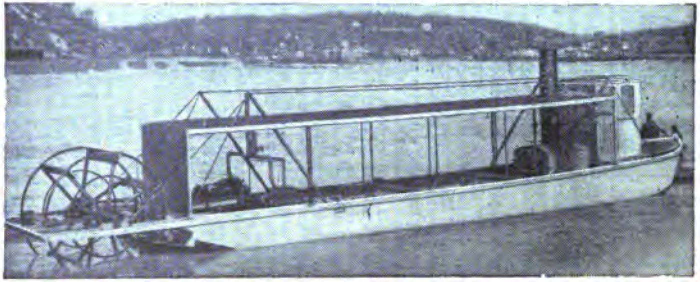


FIG. 2,057.—Light draught working boat with Seabury stern wheel engines and water tube boiler, illustrating the placement of **machinery in boats of this type.**

Stern Wheel Steam Boat Proportions

Size	Length of hull feet	Length over all feet	Beam of hull feet	Width on deck feet	Draft unloaded inches	Capacity		Towing capacity	Approx. fuel consumption ten (10) hours		Double engines	
						Tons	Draft inches		Soft coal tons	Wood cords	Diam. of cylinder and stroke	Indicated horse power 150 lbs. pressure
A	50	59	14	15½	12	10	20	30	1	2	5×20	30
B	65	76½	15	17	12	17	20	55	1¼	2¼	6×30	45
C	70	82½	16	18	14	20	21	80	1½	2½	7×32	62
D	80	94	18	20	15	25	24	110	2	3½	8×36	75
E	90	103	22	25	16	30	24	150	2½	4	9×42	111
F	110	125	25	28	18	50	28	225	3	5	10×48	157
G	135	154	30	33	22	100	32	300	4	7	12×60	214

NOTE.—The following from James Reese & Sons Co. of Pittsburgh gives some interesting data on Western river steam boat practice: "The machinery used on the boats is mostly of the high pressure type, especially on the smaller boats, as it has been found more practical, economical and simple in construction, with all types of valve motion, from the slide valve, balance slide valve, slide valve with poppet cut off, piston valve with poppet cut off, piston valve with the slide or piston cut off valve on the top or side, and the piston valve with the variable cut off valve working within the main valve, the rotary valve in center or at each end of cylinder, the lever poppet valve with balance piston to same, known as the Frisbee or Moore valve, to the double balance poppet valve; the lever poppet valve is considered the most economical, durable and simple valve motion that can be constructed on engines of the larger type. One has only to glance at the rise and fall of the levers to see if the proper valve motion be given to the engines, and with the adjustable cut off and inside cam motion has been found practically, in comparison with all other valves and valve motions, to be the very best that can be put on steamers for river navigation. The very many improvements which have been made on steam vessels originated on the stern wheel boats. ***In years past,*** the boiler feed pump was attached to the engines, and had to be disconnected when not pumping water to the boilers; then came the horizontal pump known as the wheel barrow pump; then came the ***Doctor,*** a vertical fly wheel boiler feed pump, having two cold and two hot water pumps with heater attached, and which is to-day in use on most all of our river steamers. The steam capstan was introduced on these boats in 1855, and the engine that was attached to the capstan and freight hoisters was thereafter called the ***nigger engine,*** so called inasmuch as it superseded the "colored deck crew" in changing from hand to steam power. The first double compound engines, known as the ***clipper engines*** were used on boats on the Ohio River in 1843. The first double or twin wheels, with one pair of engines to each wheel were used on stern wheel boats on the Ohio River in 1853. The balance rudders were patented and first used on our stern wheel boats in 1855 and 1856. Boats with rudders at bow and stern on stern wheel, side wheel, twin hull or catamaran with wheel in center, and the stern wheel working in recess in the hull were all in use on our rivers between 1840 and 1855. With the advent of the propeller, the propellers working at the stern, at the bow, and in recess in hull or tunnel in bottom of the boat as well, it may be truthfully said steam vessels of all known types have been operated on our rivers prior to 1860. Our small tow boats operating on the Monongahela River transport coal between given points for four cents per ton; while the rate by rail between the same points is forty-five cents per ton. And from Pittsburgh to New Orleans, a distance of two thousand miles or more, coal is transported for less than one dollar per ton. It has been practically demonstrated and cannot be denied that the stern wheel steamers engaged on our Western and Southern rivers in the passenger, freight and towing traffic have no equal for draught of water when light or when loaded, their superiority in handling, or speed when loaded to cargo capacity.

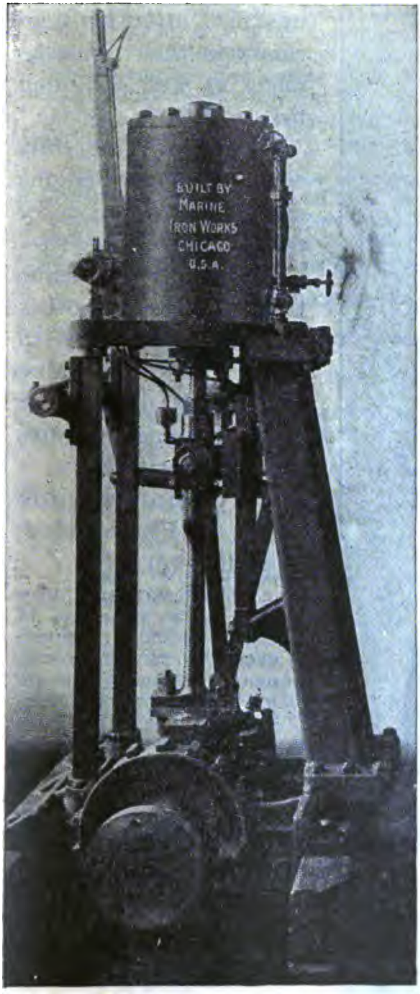


FIG. 2,058.—Marine Iron Works open front single cylinder tug engine, rear view; built in sizes 8×10 to 16×16.

Vertical Inverted Engines. — Most marine engines belong to this general class in which the cylinder or cylinders are directly over the shaft. They may be simple, compound, triple, or quadruple expansion according to the degree of economy required.

Simple Engines.—Where simplicity, durability, and large power for the space occupied by the engine are the chief considerations rather than economy, a single cylinder engine is permissible.

Before the gas engine came into general use, single cylinder steam engines were largely used on small and medium size launches, where only moderate speed was required, their popularity being due to durability, the little and comparatively unskilled attention necessary in their management as well as small first cost.

At present, the only single cylinder engine extensively used is of considerably larger size, as is found on some tugs, lighters, and other working boats. With respect to the valve gear and degree of economy, single engines may be classified as:

1. Ordinary slide valve; 2. Piston valve; 3. Riding cut off.

Notwithstanding the very poor economy of single cylinder engines, even when run condensing, many tugs and lighters are to be seen running non-condensing, the large clouds of "steam" visible and the labored puffings are eloquent evidence, of willful waste, or a non-consideration of the saving due to condensing.

The arguments (if any) usually advanced to justify high pressure

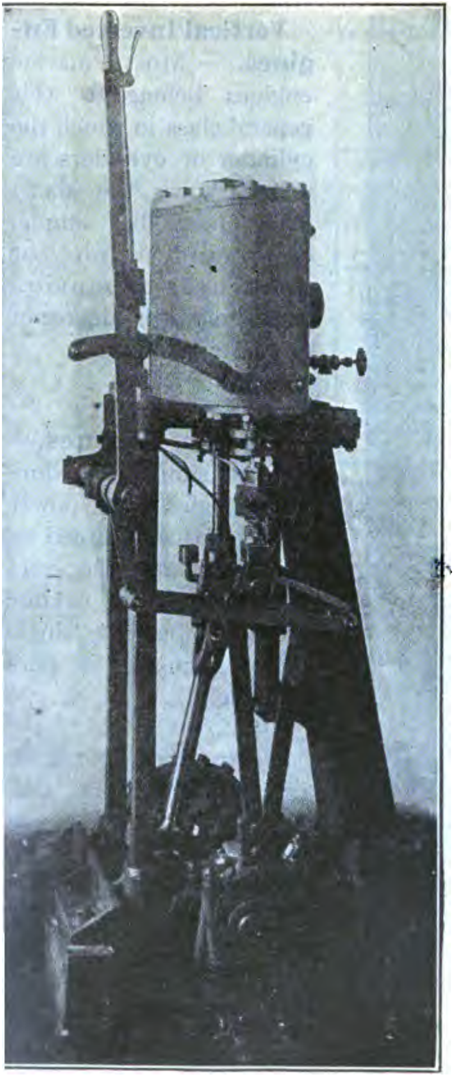


FIG. 2,059.—Marine Iron Works open front single cylinder marine engine; forward view. The end suspension of link gives minimum slip of block and the double radius rods, prevent lateral or twisting strain.

operation, are: short runs, long stand by periods, and frequent manœuvring around docks, as well as, lower grade labor, and freedom from oil in the boiler. Notwithstanding, all these items the author has always held that the practice of operating single cylinder marine engines non-condensing is not justified under any condition unless fuel is to be had at an extremely low cost. If the boat is to navigate contaminated waters unfit to pass through a surface condenser, a jet condenser could be used.

It would be a very poorly proportioned plant that would not show 15 per cent. saving by installing a condenser, and to illustrate, suppose a lighter with a 500 horse power non-condensing engine required 45 lbs. of feed water per horse power hour and the evaporation, from and at 212°, was 8 to 1, then the coal required per hour while running would be

$$500 \times 45 \div 8 = 2{,}813 \text{ lbs.}$$

The coal saved by using a condenser would be

$$2{,}813 \times .15 = 422 \text{ lbs. per hour.}$$

Assuming the boat was operated only five hours per day then the saving per working year of 300 days would be

$$422 \times 5 \times 300 \div 2{,}000 = 316\tfrac{1}{2} \text{ tons}$$

which, with coal at, say, $4.00 per ton, would be

$$316\tfrac{1}{2} \times 4 = \$1{,}266.$$

These working boats probably average more than five hours per day and it is safe to say that the saving during the first year would more than pay for the cost of the condensing apparatus, up-keep, and extra expense

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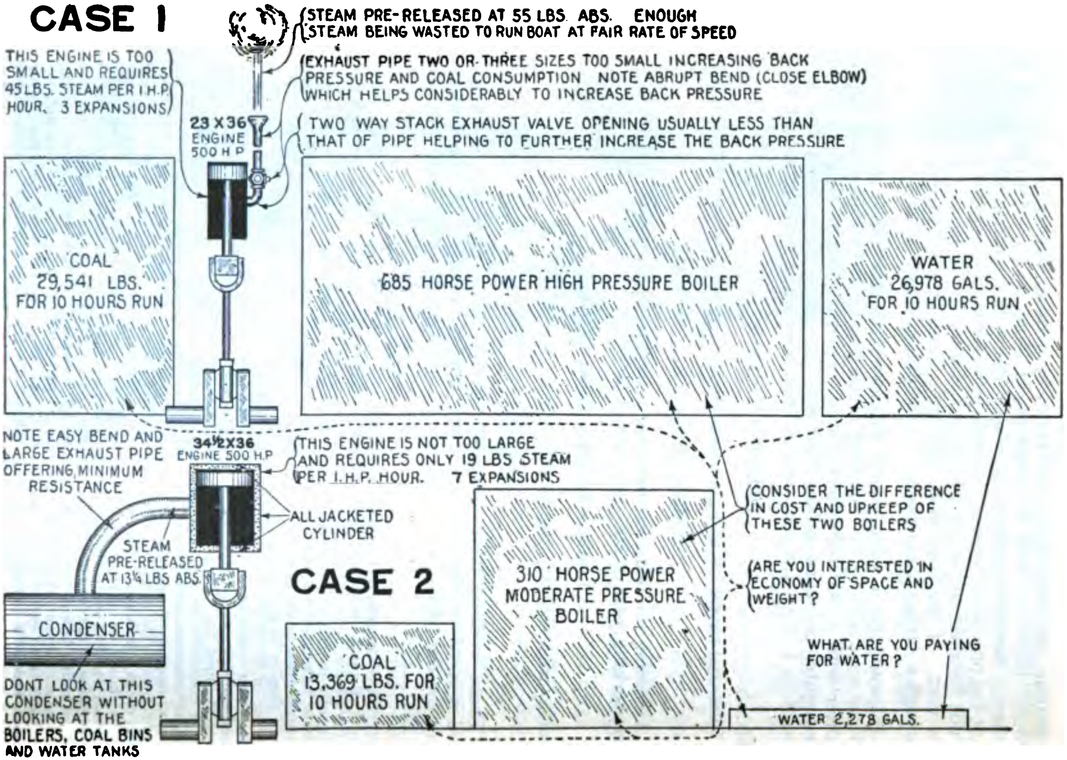
NOTE.—***Counterfeit marine engines.*** The following quotation from the Manufacturers' Record, Baltimore, Md., though relating to small engines (4 to 10 inch cylinders) contains advice, some of which may be considered in the purchase of larger engines: "It not infrequently happens that ordinary 'manufactured' vertical stationary engines are stripped of their base, balance wheel pulley, governor, etc., and with a reversing gear substituted are offered the unsuspecting buyer as a 'Marine Engine,' although for obvious reasons this does not apply to large sizes or to experienced buyers. The 'counterfeits' are easily detected, even from an engraving or from the usual details of construction that any buyer can fairly insist upon having when ordering. For instance, 'bottle' or similar shape frames, with crankshafts propped up high, rendering it difficult (size considered) to fasten the engine firmly in position or keep it 'lined up,' and practically impossible to prevent excessive vibration; while as to accurate counterbalancing, suitable steam passages for high pressure and resultant piston travel, adjustments for taking up the wear, etc., they all appear to be an unknown quantity. That such machinery, though short lived, is troublesome during its existence, is manifest. Next in importance to the design of a genuine marine engine is the proportions of the working parts, such as crank, crank pin, connecting rod, etc. All wear on reversing link, link block and link knuckles should be adjustable, every one of these adjustments being of value to the user. Additional value is gained by having the quadrant made double, with the locomotive reverse lever working in the center, attached by double connection on each side of the link block, and thus preventing lateral strain. Although a customer may in good faith accept (along with a low price) a statement that 'those little features are of no benefit except to talk on' the engine which lacks them enters a noisy protest before its first season closes and earnestly begs to be run slow, just at a period, by the way, when its 'well put up' neighbor is doing double duty without a murmur and making that little saving in first cost appear somewhat irritating."


CASE 1
STEAM PRE-RELEASED AT 55 LBS. ABS. ENOUGH STEAM BEING WASTED TO RUN BOAT AT FAIR RATE OF SPEED
THIS ENGINE IS TOO SMALL AND REQUIRES 45 LBS. STEAM PER I.H.P. HOUR. 3 EXPANSIONS
EXHAUST PIPE TWO OR THREE SIZES TOO SMALL INCREASING BACK PRESSURE AND COAL CONSUMPTION NOTE ABRUPT BEND (CLOSE ELBOW) WHICH HELPS CONSIDERABLY TO INCREASE BACK PRESSURE
23 X 36 ENGINE 500 H P
TWO WAY STACK EXHAUST VALVE OPENING USUALLY LESS THAN THAT OF PIPE HELPING TO FURTHER INCREASE THE BACK PRESSURE
COAL 29,541 LBS. FOR 10 HOURS RUN
685 HORSE POWER HIGH PRESSURE BOILER
WATER 26,978 GALS. FOR 10 HOURS RUN
NOTE EASY BEND AND LARGE EXHAUST PIPE OFFERING MINIMUM RESISTANCE
34½X36 ENGINE 500 H.P.
THIS ENGINE IS NOT TOO LARGE AND REQUIRES ONLY 19 LBS STEAM PER I.H.P. HOUR. 7 EXPANSIONS
ALL JACKETED CYLINDER
STEAM PRE-RELEASED AT 13¼ LBS ABS.
CASE 2
CONSIDER THE DIFFERENCE IN COST AND UPKEEP OF THESE TWO BOILERS
ARE YOU INTERESTED IN ECONOMY OF SPACE AND WEIGHT?
CONDENSER
310 HORSE POWER MODERATE PRESSURE BOILER
WHAT ARE YOU PAYING FOR WATER?
DONT LOOK AT THIS CONDENSER WITHOUT LOOKING AT THE BOILERS, COAL BINS AND WATER TANKS
COAL 13,369 LBS. FOR 10 HOURS RUN
WATER 2,278 GALS.

FIGS. 2,060 and 2,061.—Two steam lighter power plants illustrating usual practice (*case 1*), and the author's system (*case 2*). See chapter 52 for engine details.

if any, for higher grade of engineer, making the practice of high pressure operation inexcusable.

A very common mistake, and one which tends to prejudice boat owners against the use of condensers, is the installation of engines too small for the load making it necessary to use initial pressures anywhere from 100 to 150 lbs. gauge. An engine operating under these conditions cutting off at one-third or one-fourth stroke will have a terminal pressure considerably above atmosphere and to obtain a 24 or 26 inch vacuum would require an abnormally large (and expensive) condenser, moreover the excessive temperature range will cause excessive condensation in the cylinder and considerably reduce the saving due to the condenser.

The author believes that 80 lbs. gauge initial pressure should be the limit for single cylinder engines and for maximum economy, there should be from 6 to 7 expansions with not less than 26 inch vacuum, using superheated steam with unjacketed cylinder or saturated steam with jacketed cylinder.

The following example will illustrate the difference in size of a single cylinder engine proportioned according to the usual practice and according to the author's views.

Example.—Find the cylinder dimensions of a 500 horse power, single cylinder, lighter engine operating under the following conditions: ***Case I.*** Initial pressure, 125 lbs.; cut off ⅓ stroke; back pressure, 3 lbs. abs. (23.8" vacuum); 110 revolutions; unjacketed cylinder, diagram factor .8. ***Case II.*** Initial pressure, 80 lbs.; cut off, $^1/_7$ stroke; back pressure, 2 lbs. abs. (25.9" vacuum); 125 revolutions; jacketed cylinder, diagram factor .9.

Case I. Usual practice

$$125+15=140 \text{ lbs. abs. initial pres.}$$

$$\text{M. E. P.} = \left(\frac{140\times 2.099}{3}-3\right)\times .8=76 \text{ lbs.}$$

For 36" stroke

Size
23 × 36

$$\text{piston area}=\frac{500}{.000004\times 36\times 110\times 76}=415 \text{ sq. ins.}$$

$$\text{piston diameter} =\sqrt{\frac{415}{.7854}}= 22.9, \text{ say, } 23''.$$

Case II. Author's proportions

$$80+15=95 \text{ lbs. init. pres.}$$

$$\text{M. E. P.} = \left(\frac{95\times 2.946}{7}-2\right)\times .9=34 \text{ lbs.}$$

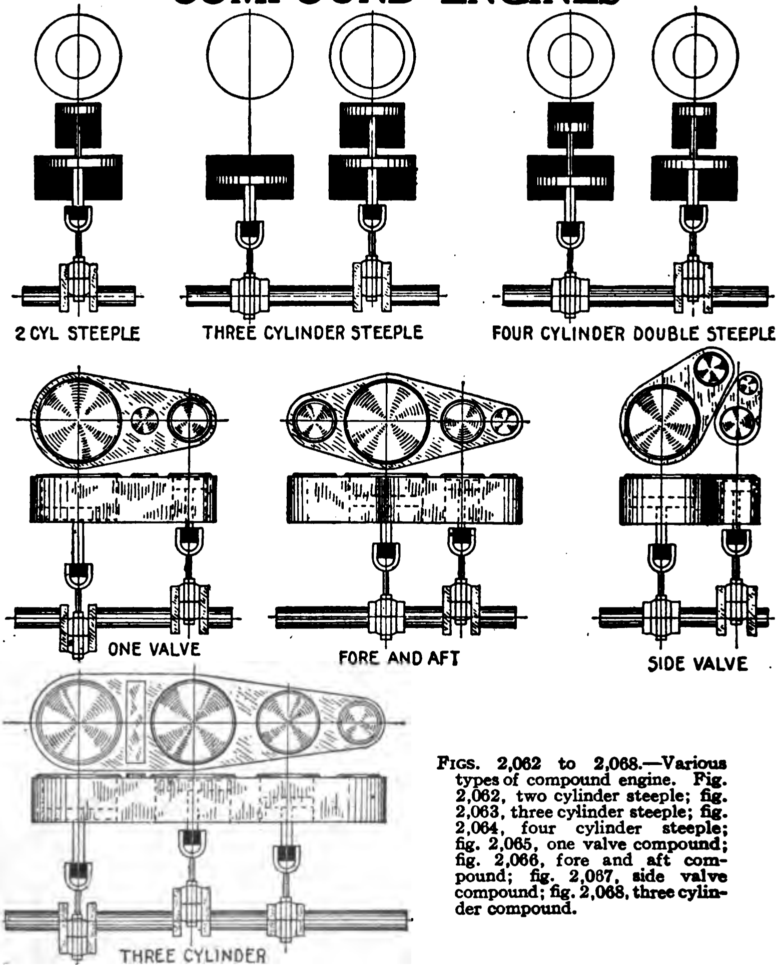
Size
34½ × 36

For 36" stroke

$$\text{piston area} =\frac{500}{.000004\times 36\times 110\times 34}=929 \text{ sq. ins.}$$

$$\text{piston diameter} =\sqrt{\frac{929}{.7854}}= 34.4, \text{ say, } 34\tfrac{1}{2}''.$$

COMPOUND ENGINES



FIGS. 2,062 to 2,068.—**Various types of compound engine. Fig. 2,062, two cylinder steeple; fig. 2,063, three cylinder steeple; fig. 2,064, four cylinder steeple; fig. 2,065, one valve compound; fig. 2,066, fore and aft compound; fig. 2,067, side valve compound; fig. 2,068, three cylinder compound.**

According to tests[*] a saving as high as 40.4 per cent has been made by the use of steam jackets[ƒ] on a single cylinder Meyer cut off engine.

Case II, furnishes ideal conditions for the use of a jacket, and having in mind the large clearance of the Meyer cut off and the effect of clearance on economy, it would seem possible to still further reduce the feed water consumption by the use of rocking valves located in the heads. In regard to clearance, it should be noted that its detrimental effect is not measured in terms of per cent of displacement of full stroke but per cent of displacement up to point of cut off, and at very short cut off, as one-seventh, this becomes considerable; accordingly with the ordinary clearance, the saving due to the short cut off is largely offset.

The author believes that the feed water consumption of the ordinary single cylinder marine engine can be reduced at least 50 per cent, with the following design and working conditions: cylinder and heads steam jacketed, four rocking valves in the heads (reducing clearance to almost zero); variable cut off by shifting eccentric; separate eccentric for exhaust valves; 80 lbs. initial pressure; 6 to 7 expansions, 26 inch vacuum; speed somewhat higher than usual practise to avoid an abnormally larger engine for the power delivered.

Compound Engines.—For moderately short runs, and for vessels requiring quick and frequent manœuvring around docks, the compound engine can sometimes be used to better advantage than the triple engine. As compared with the latter compound engines cost less, are easier to handle, and less boiler pressure is required to secure good results. There are several types of compound or two stage expansion engines, which may be classed according to the arrangement of the cylinder, as

1. Steeple
   - two cylinder
   - three “
   - four “
2. Fore and aft;
3. Side valve;
4. Three cylinder.

---

[*] NOTE.—Special attention is called to these tests, which were made by Bryan Donkin, and described in the note on page 108; those interested should read the report of the research committee on the value of steam jackets comprising an elaborate series of experiments (see *Pro. Inst. Mech. Eng.* 1892, page 464). ***It should be noted,*** as stated above, that the steam jacket saving of 40.4% was obtained with an engine fitted with the Meyer cut off gear—a gear not adapted to obtaining best results possible with a very early cut off because of the large clearance. This large clearance may be reduced to almost zero by the four valve engine shown in fig. 2,060. As shown, a shifting eccentric operates the admission valves, and a fixed eccentric the exhaust valves. With this arrangement the best economy possible for a given cut and temperature range in a single cylinder counter flow engine with non-releasing valve gear should be obtained.

[ƒ] NOTE.—The diversity of opinion which still exists as to the value of the steam jacket is due to its mis-application, the tests in such cases being misleading except to the better informed. A jacket should only be used with a very short cut off, and to obtain the full economy the clearance, as mentioned above, should be reduced to a minimum. For maximum effect heads and piston as well as the cylinder should be jacketed and ***proper provision for drainage made.*** See Prof. Prosser's tests.

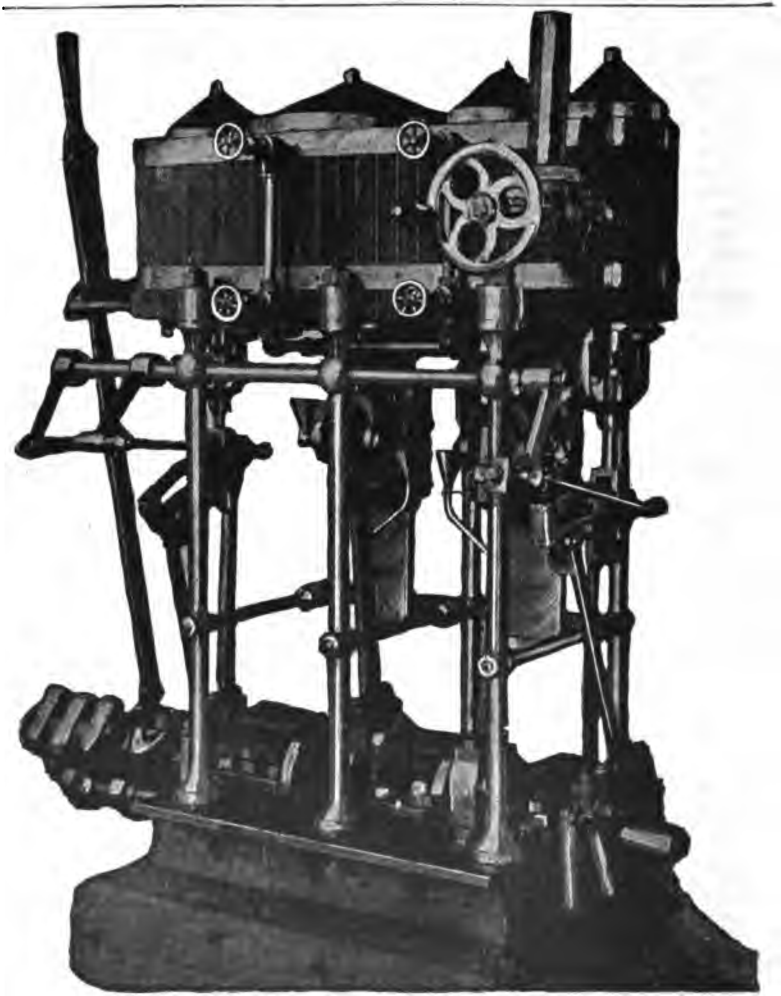


Fig. 2,069.—Fore and aft compound engine, designed by the author for 35′×7′ launch *"Stornoway"*; illustrating the accessibility due to the open column type frame; also, self-contained ball thrust bearing. Cylinders 3½, and 8, by 6; cylinder ratio 1 : 5.23; shifting link motion with adjustable arm on *h.p.* gear; piston valves; mahogany cylinder lagging with nickel plated bands and cylinder head bonnets. The bearings are proportioned for high speed heavy load service.

For small and medium size steam craft, especially for hard working boats where economy of space is of *extra* value, a two cylinder steeple or vertical tandem compound engine is often desirable, as shown in fig. 2,070.

Two other possible arrangements are shown in figs. 2,060 and 2,061; with the present boiler pressure such construction does not represent good practice, but was permissible in the early days of low boiler pressures in the case of very large powers.

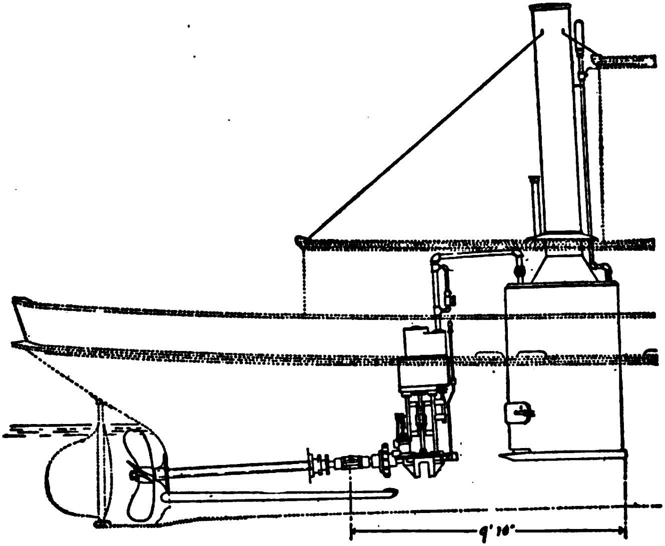
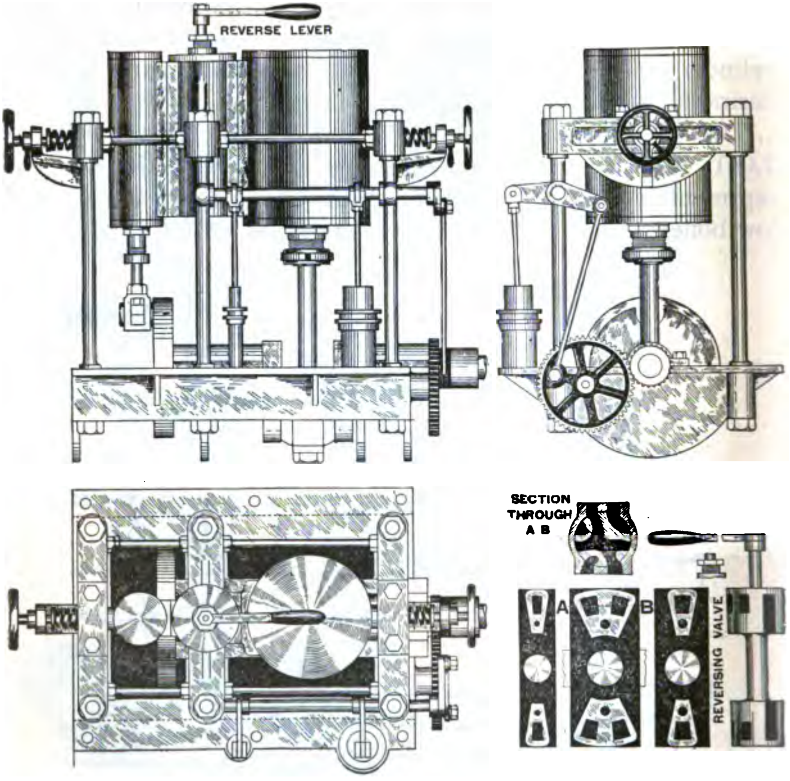


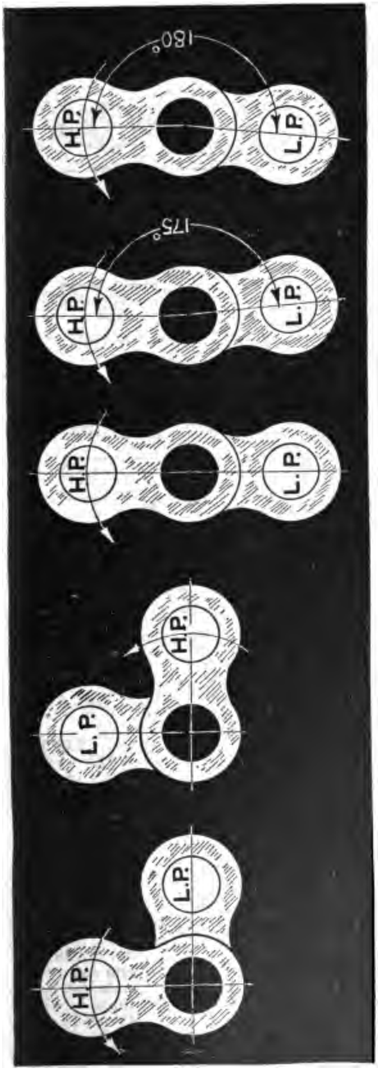
FIG. 2,070.—View showing arrangement of Marine Iron Works heavy duty steeple compound and vertical boiler in working vessel. The dimension shows the space occupied by a 50 *h. p.* outfit.

Figs. 2,062, 2,065 to 2,067 show four types of two cylinder compounds. In fig. 2,065, only one valve is used making it necessary that the cranks be at 180° or nearly so. The arrangement gives simplicity and better balance than 90° and is sometimes preferred (especially in the small and medium sizes) for these reasons, but has the disadvantage of dead centers. To overcome this the crank angles of 175° or 185° are sometimes used, the balance of moving parts being practically as good as with the 180° angle. The most popular type of compound engine especially



FIGS. 2,071 to 2,079.—Graham special two-cylinder, double acting, transfer expansion, jacketted, oscillating, marine engine, with feed and air pumps attached and ball thrust bearing. Fig. 2,071, elevation; fig. 2,072, aft end view; fig. 2,073, plan; fig. 2,074, section of reversing valve through AB; fig. 2,075, transfer cylinder ports; fig. 2,076, valve seat expansion side; fig. 2,077, expansion cylinder ports; fig. 2,078, reversing valve stuffing box; fig. 2,079, reversing valve. The engine is reversed by turning the reversing valve lever about 90°. Owing to the high speed of the engine the pumps are geared down as shown in fig. 2,072. The arrangement makes a very light and unique engine for a small high speed boat, and one that does *not* require superheated steam, being designed for *saturated* steam of moderate pressure. The 1½ and 4 by 6 in. stroke size at 1,000 *r.p.m.* (or the 1½ and 4 by 8 in. size at 750 *r.p.m.*) will develop about 15 horse power with 100 lbs. boiler pressure.

NOTE.—***Working Drawings*** (blue prints) with all dimensions necessary to build the above engine may be obtained from the publishers.



FIGS. 2,080 to 2,084.—Sequence of cranks in two cylinder compound engines. Fig. 2,080, 90° h. p. crank leading; fig. 2,081, 90° . p. crank leading; fig. 2,082, 180°; fig. 2,083, 175°; fig. 2,084, 185°.

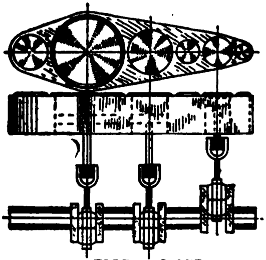
for small and medium powers is the fore and aft engine shown in fig. 2,066, with cranks at 90°. The valves being at the ends gives an amply large receiver.

In order to economize fore and aft space the side valve arrangement shown in fig. 2,067 is used. This requires toothed gears to drive the valve shaft and unless of good design and well constructed is liable to give trouble. The method of reverse is shown in figs. 543 to 549. Another objection is the large clearance in the *h. p.* cylinder necessary to bring the valves in line with the valve shaft when the latter is parallel with the main shaft. However, this type has been very extensively used especially on yachts.

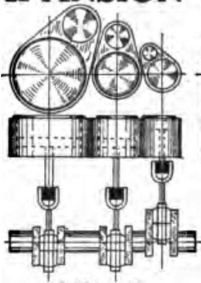
Fig. 2,068 shows a three cylinder compound with the two *l. p.* cranks at 180° to each other and at 90° with the *h. p.* crank; this arrangement, though quite extensively used, even in medium powers, is open to criticism and the author does not see the wisdom of such construction under any conditions.

Triple Expansion Engines.—This form of marine engine is more extensively used than any other for medium and large powers, as for coast and ocean steamers. Triple expansion

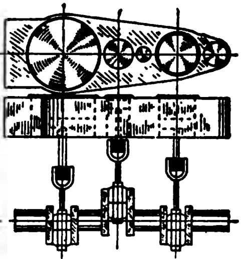
TRIPLE EXPANSION ENGINES



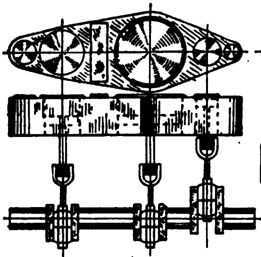
FORE AND AFT



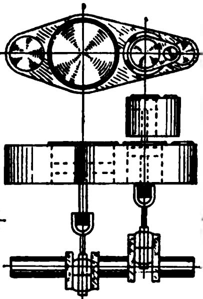
SIDE VALVE



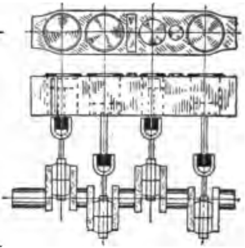
INT. AND L.P. CYLS. AT ENDS



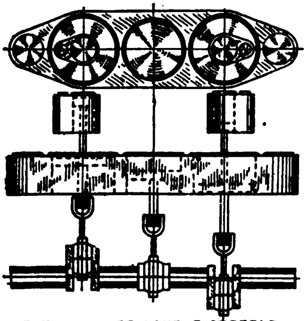
H.P. AND INT. CYLS. AT ENDS



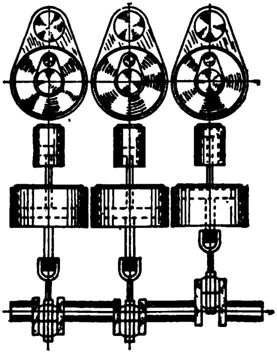
THREE CYLINDER STEEPLE



FOUR CYLINDER



FIVE CYLINDER DOUBLE STEEPLE



SIX CYLINDER TRIPLE STEEPLE

Figs. 2,085 to 2,092.—Various types of *triple expansion engine.* Fig. 2,485 natural sequence of cylinders; fig. 2,086, same sequence with side valves; fig. 2,087, *int.* and *l.p.* cyls. at ends; fig. 2,088, *h.p.* and *int.* cyls. at ends; fig. 2,089, three cylinder steeple; fig. 2,090, four cyl. with *l.p.* cyls. at ends; fig. 2,091, five cyl. steeple; fig. 2,092, six cyl. steeple with side valves.

engines are made in an unnecessarily great variety of cylinder arrangement as shown in figs. 2,085 to 2,092.

Fig. 2,085 shows the natural sequence of cylinders: *h.p.*, *int.* and *l.p.* This is the most used type, steam going directly from one cylinder to another. Frequently the cylinders are arranged in this sequence with side valves, as in fig. 2,086, giving the minimum piping between cylinders.

Figs. 2,087 and 2,088 show two cylinder arrangements which require more piping and possess no advantages over the natural sequence shown in fig. 2,085.

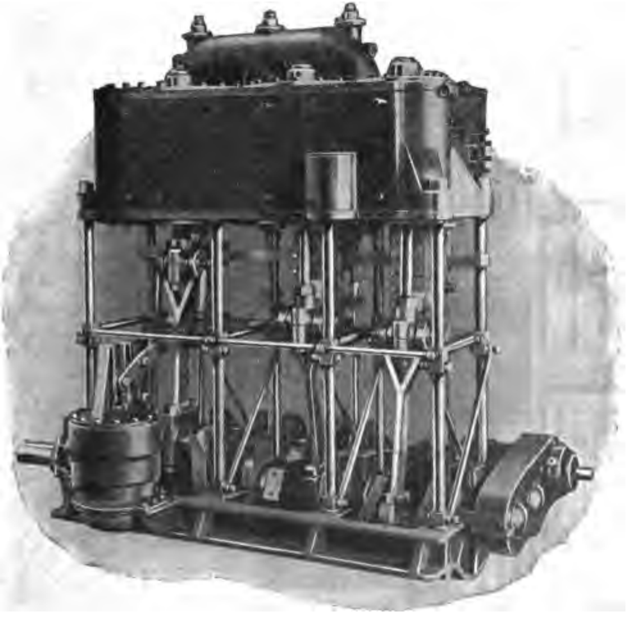


FIG. 2,093.—Seabury 275 horse power three cylinder triple expansion yacht engine, size 8×12¾ and 20×10.

Fig. 2,089 shows a three cylinder steeple, a questionable arrangement. The usual arrangement of four cylinder triple is shown in fig. 2,090, the two *l.p.* cylinders being at the ends, the objects in view being to improve the balancing, and for large power to avoid a very large casting for the *l.p.* cylinder.

Fig. 2,091 shows a five cylinder steeple arrangement for high power and fig. 2,092 a six cylinder steeple engine for exceptionally large power. In this engine there are three *l.p.* cylinders and two intermediates as shown, to avoid very large castings.

The following example will illustrate the graphical method of finding the cylinder dimensions for a triple expansion engine.

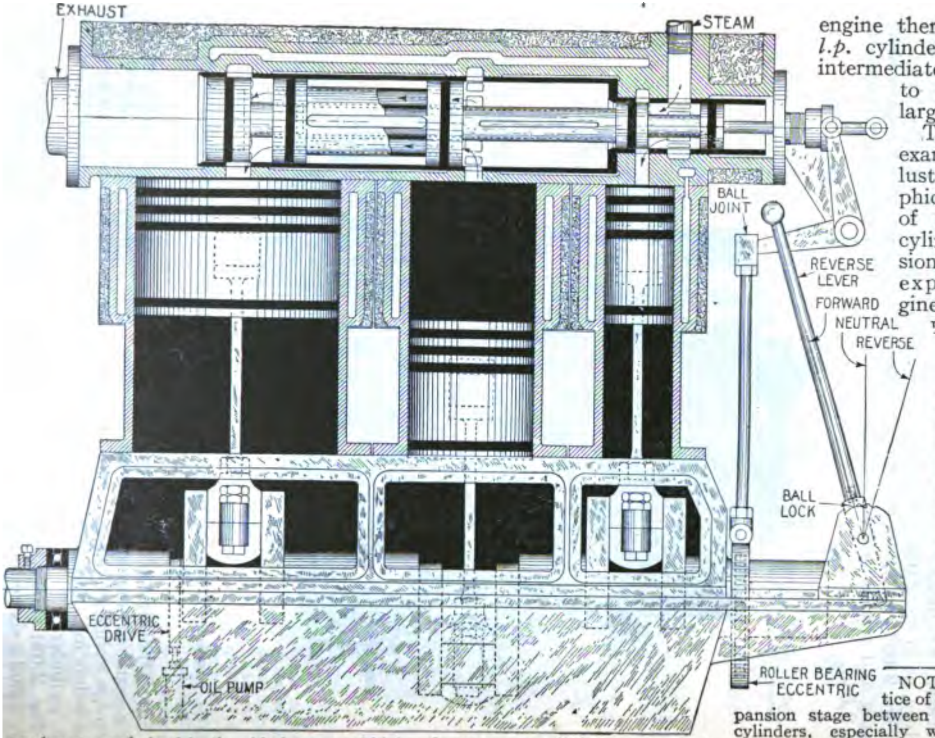


FIG. 2,094.—Graham high speed single acting, jacketted triple expansion engine. The cylinder ratio is unusually high in order to obtain full benefit from jackets. The simplicity of the valve gear is apparent, one valve only being used. Engine suitable for high speed boats.

NOTE.—The practice of dividing an expansion stage between two or more cylinders, especially when separate cranks are used, is objectionable because of higher first cost, no gain in economy but a decrease in mechanical efficiency, larger space required, more machinery to look after, etc. ***In general,*** the number of expansion stages should be increased rather than the number of cylinders and reciprocating parts per stage, in order that the increased outlay will be justified by the saving in fuel.

The engine is of the same horse power as the single cylinder engine in the previous example, but is intended for a very different kind of service.

Example.—Find the cylinder dimensions of a high speed triple expansion engine for a passenger steamer to run as follows: initial pressure 160 lbs.; number of expansions, 16; 2 lbs. back pressure; 140 revolutions per minute.

The graphical method consists in drawing a theoretical diagram from the given data of initial pressure, total number of expansions, etc., and inscribing within, diagrams for the various cylinders of such contour as will represent the expected actual performance of the engine. It is evident, then, that the success of this method depends upon the experience and judgment of the designer in drawing these "expected diagrams," their forms being obtained by first examining a large number of indicator diagrams from engines similar to the one to be designed and running under the same conditions with open column frame, and built up bed plate.

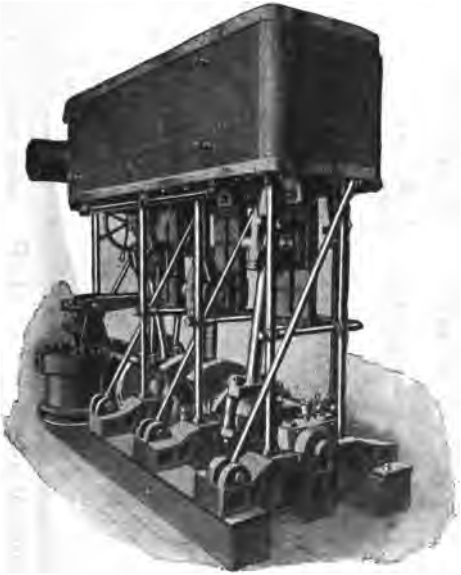


Fig. 2,095.—Seabury 600 horse power 12 inch stroke engine, with air pump driven from low pressure cylinder cross head.

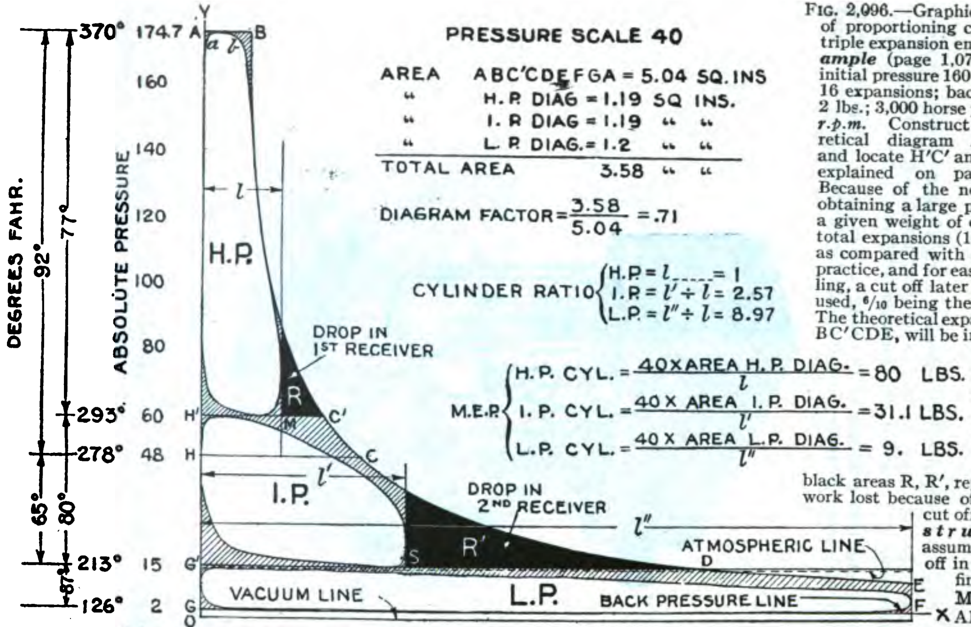


FIG. 2,096.—Graphical method of proportioning cylinders in triple expansion engines. ***Example*** (page 1,079): Given initial pressure 160 lbs. gauge; 16 expansions; back pressure 2 lbs.; 3,000 horse power; 140 *r.p.m.* Construct the theoretical diagram ABCEFG, and locate H'C' and G'D as explained on page 1,081. Because of the necessity of obtaining a large power with a given weight of engine, the total expansions (16) are low as compared with stationary practice, and for ease of handling, a cut off later than ½ is used, 6/10 being the standard. The theoretical expansion line BC'CDE, will be interrupted by ***drop*** or free expansion in the receivers giving the triangular solid black areas R, R', representing work lost because of the late cut off. ***In construction***, assuming 6/10 cut off in *h. p. cyl.*, find point M, such that AB = 6/10 of its distance *l*, from OY, and draw expected card. For equal cylinder diameters *l*, = stroke of *h.p. cyl.*, but for equal strokes, *l*, represents relatively area of *h.p. piston*. The theoretical volume exhausted into *i. p. cyl.*, is represented by H'C'. Similarly with 6/10 cut off, take S, at a distance *l'*, from OY, such that H'C' = 6/10 *l'*, and draw *expected card*, then *l'*, with equal strokes, represents relatively, area of *i. p. piston*. The relative size of the low pressure piston is represented by *l''*. The *l.p.* expected card is now drawn assuming a cut off (within limits) that will give most desirable receiver pressure. Calling *l*, unity, then *l*, *l'*, and *l''*, represent the ***cylinder ratio***. By measurement *l*, *l'*, and *l''* = .5921 1.53, and 5.34 ins. repectively, and calling *l*, unity, the cylinder ratio = 1, 2.57 and 8.97, as tabulated above. The areas of the cards are preferably obtained by planimeter; these and the *m.e.p.* for each cylinder are also tabulated above. See accompanying text.

In fig. 2,096 lay off the axes OY, of pressures and OX, volumes. On OX, take any convenient distance l'' and divide it into 16 parts—the total number of expansions.

Locate A, at a height equal to the initial pressure or $160+14.7=174.7$ lbs. absolute. Since there are 16 expansions draw $AB=\frac{1}{16}$ of l'', and describe the hyperbolic expansion curve BC′CDE. Draw GF, the 2 lbs. back pressure line thus completing the theoretical diagram ABC′CDEFGA which is the total theoretical work area of the engine.

It is desirable, in order that the stresses may be equally divided, that the cylinders develop equal power. Hence, with planimeter, locate by trial the lines HC and G′D, so that they divide the diagram into three equal areas ABCHA, HCDG′H, and G′DEFGG′.

By measurement, the pressures corresponding to G′ and H, are 15 lbs. and 48 lbs., these are placed to the left of the vertical axis of pressures and in the second column the temperatures corresponding to these pressures.

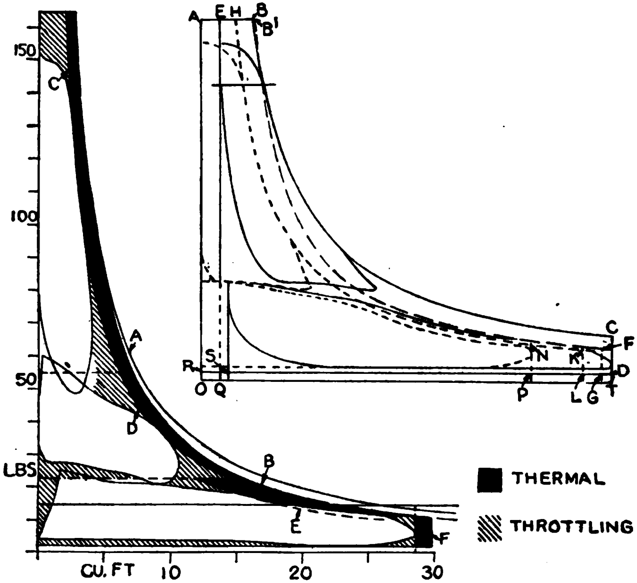
For equal theoretical areas then the range of temperatures in the different cylinders is: *h. p.* 92°; *i. p.* 65°; *l. p.* 87°.

Since the main object of multi-cylinders is to reduce the loss due to condensation by reducing the range of temperature in any one cylinder, best economy then will be secured by equalizing the temperatures obtained above.

Equal distribution of work and equal temperature ranges being opposing factors, the area may be modified to secure only approximately equal distribution of work. In the diagram, the greatest difference in ranges is between the *h. p.* and *i. p. cylinders*, hence taking an average of these by moving the line HC, up to H′C′, the ranges will be found very nearly equal in ali cylinders, giving the theoretical areas ABC′H′A, H′C′DG′H′ and G′DEFGG′.

The next step is to find the ***cylinder ratio*** and sketch in the ***"expected diagrams,"*** as explained in fig. 2096 which, as mentioned are made to conform with indicator cards from similar engines operating under similar conditions. These expected diagrams are the white areas marked H P., I. P., and L. P., the solid black areas, representing the ***drop*** losses and the cross sectioned areas, those due to condensation and friction of the steam in passing through the valves and passages.

At this stage it is well to check the judgment used in drawing the expected diagrams by finding the ***diagram factor*** and comparing this with diagram factors of similar engines already built and tested. This factor is *the ratio between the sum of the areas of the expected cards and the area of the theoretical diagram*, and is, as tabulated in the figure, .71. Here it should be noted that the diagram factor for triple expansion engines may be as low as .6, depending upon the type of engine and operation, hence, in actual design, if a diagram factor less than .71 were aimed at, the expected cards as drawn in fig. 2,096 would be cut down until the desired factor was obtained. In such procedure, it would help to obtain diagram factors also for each cylinder and compare them with known factors from other engines in operation. Because of the late cut off used on marine engines the expansion is not continuous but is interrupted by *drop* at release, or sudden fall of pressure due to *free expansion* in the receivers.



FIGS. 2,097 and 2,098.—Combined diagrams illustrating the diagram factor in multi-stage expansion engines. Prof. Heck says: "One method is to take the ordinary combined diagram with the clearance line as vertical axis, and draw the hyperbola to touch the *h. p.* expansion curve at cut off, extending it from the boiler pressure to the end of the *l. p.* diagram. In fig .2,098, this makes ABCDR, the ideal figure, though some engineers have used the base line OT, instead of the condenser pressure line RD, as the bottom of the diagram. Using RD, the area of the indicator diagrams is 57.2 per cent of the area of ABCDR. Busby gives a collection of factors which lie in the neighborhood of .67 ranging from .6 to .75 for various classes of marine engine. The really logical scheme is to transfer the diagrams into the dotted position and make the compression curves agree with the axis AO, as in fig. 2,097 (which is from the S. S. Meteor referred to compression curves). When this is done the hyperbola HN, is drawn, with O, as origin, and the effective combined area is compared with AHNPR, coming here to 84.7 per cent. This method involves too much graphical work and changes the lengths of the respective diagrams to something else than the actual cylinder volumes (as laid off to scale). For convenient practical use the best method is to bring the several end lines to a common axis, as would be done if the *l. p.* diagram were moved a little way to the left in fig. 2,098 so as to touch EQ. Thus, with Q, as the origin, the hyperbola B'F, is drawn, to touch the *h. p.* expansion curve at a point on a horizontal line through the upper end of the compressure curve—this making the diagram QEB'F, have the same width, at any pressure, as OAHN. The length S G, is the same as that of the *l. p.* diagram, representing, as is proper, the volume of the large cylinders. With this axis, EB'FGS, as standard, the factor is .806. Variation in the standard used by different writers makes the diagram factor method less useful than it might be. Even for the roughest preliminary work it is better to sketch out the diagram and measure it."

In the diagram, the two solid black triangular areas show the extent of this loss, and by drawing a number of diagrams for different working conditions it will be noted that the loss by drop increases as the initial pressure and number of expansions are increased, thus limiting the range for best economy in three stage expansion with drop.

Assuming the expected diagrams in fig. 2,096 to represent approximately the actual diagrams, the cylinder dimensions are found as follows:

Distribution of power

Total area expected diags. = 1.19 + 1.19 + 1.2 = 3.58 sq. in.

Power h. p. cyl = 3.000 × 1.19 ÷ 3.58 = 997 horse power
" i p. " = " × " ÷ " = 997 " "
" l. p. " = 3,000 × 1.2 ÷ 3.58 = 1,006 " "
Total 3,000 " "

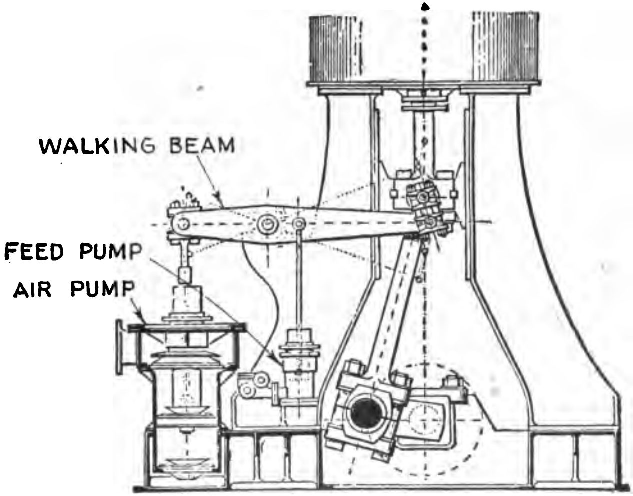


Fig. 2,099.—Sectional view ot engine showing connection of air and feed pumps direct to cross head of the engine. This is the approved method of driving these pumps when the engine is of such size that its speed is not beyond the speed limit of the pumps. The drive consists of a system of rocking levers and connections as shown. The features of engine driven pumps are the absence of all steam cylinders with their working parts and the great saving in attendance and economy and avoidance of uncertainty of operation.

Stroke

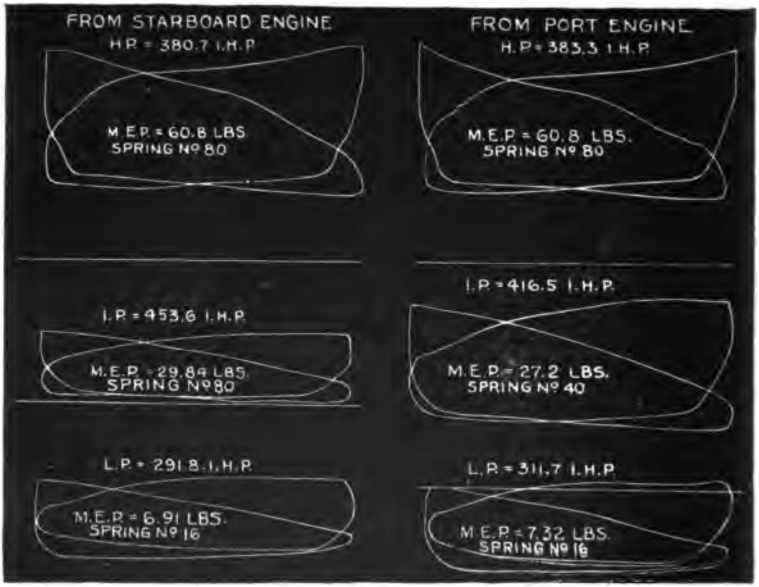
The length of stroke should be such that will give a proper piston speed for the service required. For 140 revolutions per minute a stroke of 3 feet will give a piston speed of

$$2 \times 3 \times 140 = 840 \text{ ft.}$$

which represents a medium speed and may be adopted, assuming that this will give a *h. p.* cylinder diam. somewhat less than the stroke.

High Pressure Cylinder

M. E. P. = as (in fig. 2,096) = 80 lbs.



Figs. 2,100 to 2,105.—Indicator cards from the triple expansion engines of the twin screw steamer *Monmouth*, C. R. R. of N. J. (Sandy Hook Route), size of engine 19¼", 30", and 50" by 30" stroke. Cards taken during a speed trial at Delaware Breakwater. Length of course 1.261 nautical miles. The data is as follows:

Starboard Engine Steam Pressures						Total I. H. P.	Time over course	Speed in knots	Speed in miles
Steam Boiler	1st receiver	2nd receiver	Vacuum	R.P.M.	I. H. P.				
155	55	7	26	142	1,126.1	2,237.6	3–58	19.07	22
Port Engine									
	56	5	26	143	1,111.5				

$$\text{area piston} = \frac{997 \times 33{,}000}{80 \times 840} = 489.59, \text{ say } 490 \text{ sq. ins.}$$

$$\text{diameter corresponding} = \sqrt{\frac{490}{.7854}} = 24.98, \text{ say } 25 \text{ ins.}$$

Intermediate Pressure Cylinder

$$\text{area piston} = \frac{997 \times 33{,}000}{31.1 \times 840} = 1{,}259.2, \text{ say } 1{,}260 \text{ sq. ins.}$$

$$\text{diameter corresponding} = \sqrt{\frac{1{,}260}{.7854}} = 40.1, \text{ say } 40 \text{ ins.}$$

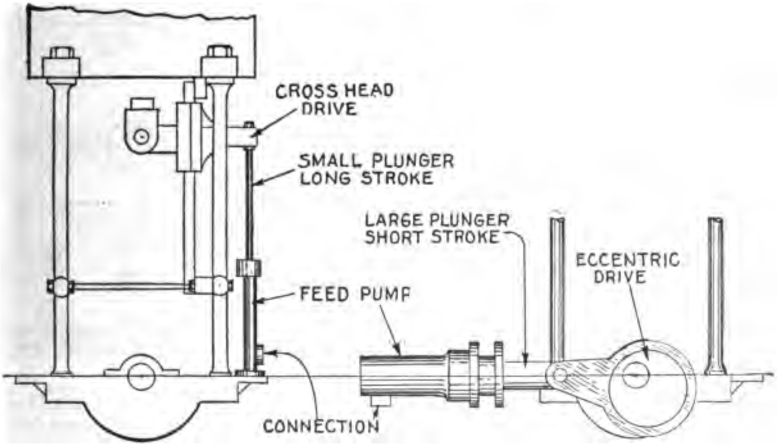


FIG. 2,106.—***Cross head drive*** for pump. The plunger is attached direct to the cross head. This is a very simple arrangement and because of the long stroke the plunger is of very small cross sectional area, thus bringing very little stress on the arm projecting from the cross head. This forms a simple and desirable arrangement when the piston speed is slow enough for the satisfactory working of the pump. The speed limit can be extended by using valves of liberal size, thus the volume of water is passed with very little lift of the valves, and the noise and jar due to seating reduced.

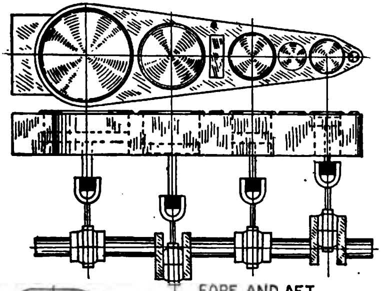
FIG. 2,107.—***Eccentric drive*** for pump. This is a satisfactory and common form of drive, though the eccentric introduces more friction than in other forms, necessitating closer attention to lubrication.

Low Pressure Cylinder

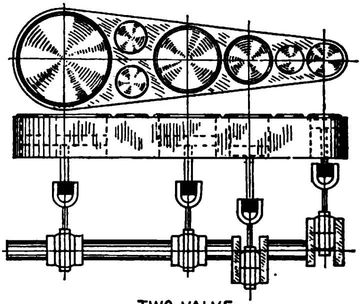
$$\text{area piston} = \frac{1{,}006 \times 33{,}000}{9 \times 840} = 4{,}391 \text{ sq. ins.}$$

$$\text{diameter corresponding} = \sqrt{\frac{4{,}391}{.7854}} = 74.76, \text{ say } 74\tfrac{3}{4} \text{ ins.}$$

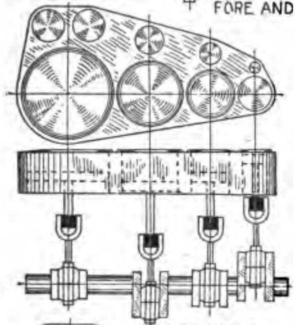
QUADRUPLE EXPANSION ENGINES



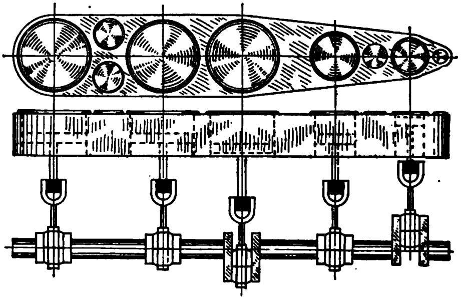
FORE AND AFT



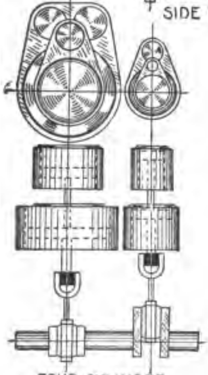
TWO VALVE



SIDE VALVE



FIVE CYLINDER



FOUR CYLINDER DOUBLE STEEPLE

FIGS. 2,108 to 2,112.—Various types of quadruple expansion engine. Fig. 2,108, four cylinder form with cranks at 90°; fig. 2,109, four cylinder, two valve with cranks at 180°, except *1st* and *2nd int*. cranks which are at 90°. This type gives a simplified valve gear; fig. 2,110, four cylinder, side valve, with double piston valve on the *l.p.* cylinder; fig. 2,111, five cylinder with cranks at 90° except the two *l.p.* cranks which are at 180° to each other; fig. 2,112, four cylinder, double steeple with cranks at 90°.

Quadruple Expansion Engines.—For four stage expansion. the boiler pressure should be 200 lbs. or more.

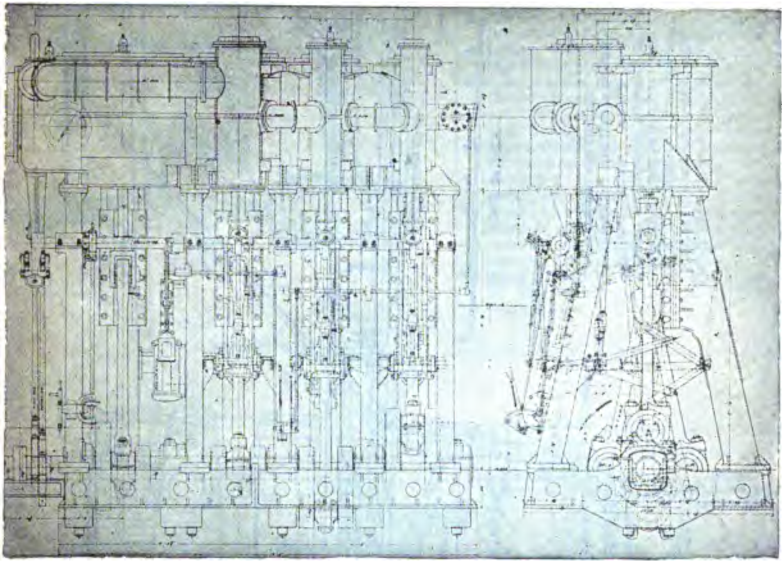


Fig. 2,113.—American Shipbuilding Company quadruple expansion engine. Cylinder, 18½—28½—43½—66×42 stroke; 2,000 *i.h.p.* The *h.p.*, first and second *int.* cyls. are fitted with piston valves, driven by Joy valve gear, while the *l.p.* cyl. has a double ported slide valve fitted with shifting link motion, the arrangement making a short and compact engine, which is necessary on the Lake freighters where fore and aft space in the engine room must be a minimum.

Even at 200 lbs. pressure the triple expansion engine with large cylinder ratio, is, notwithstanding the increased loss due to drop or free expansion in the receivers, almost as economical as the quadruple engine and is, of course, less complicated.

The many cylinders of a quadruple expansion engine make it possible to use many combinations. The most natural arrangement is four cylinders in the sequence high, first intermediate, second intermediate, low pressure, the cranks being placed ninety degrees as in fig. 2,108, some other arrangements being shown in figs. 2,109 to 2,112.

Fig. 2,112 shows a double steeple arrangement requiring only two cranks. This arrangement gives the minimum flow space. The cylinder dimensions of a quadruple engine may be calculated by the graphical method similar to the construction in fig. 2,096, dividing the diagram into four areas instead of three, or by calculation only, using the diagram factor.

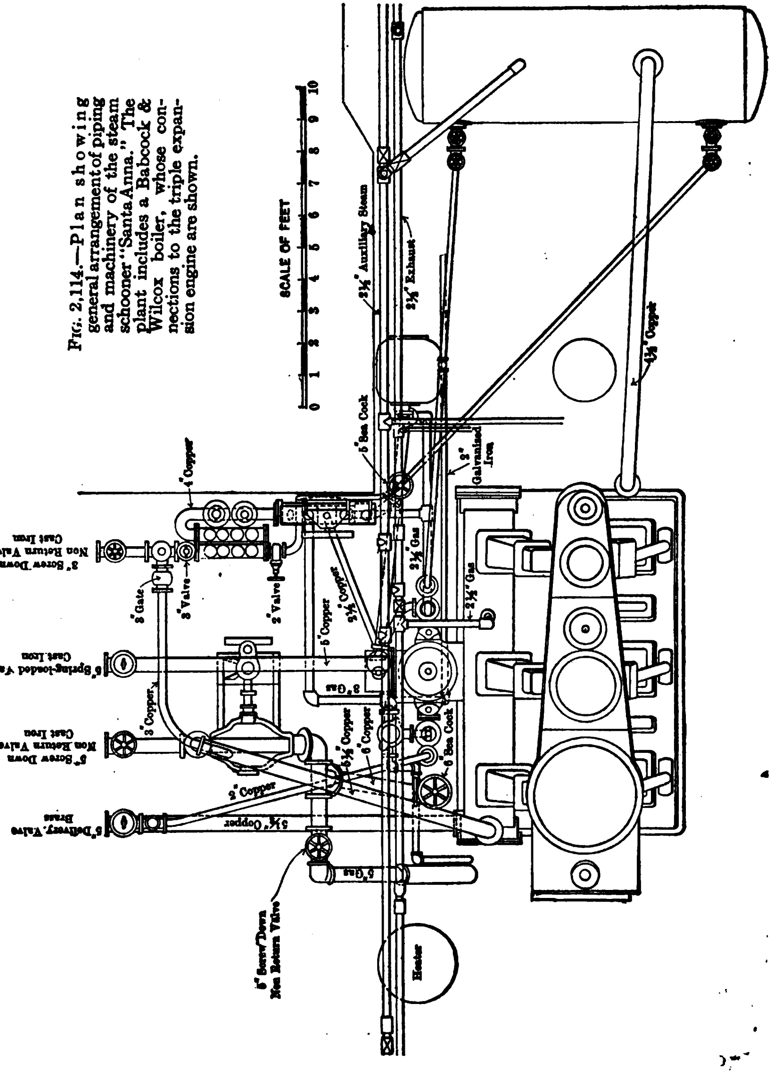


FIG. 2,114.—Plan showing general arrangement of piping and machinery of the steam schooner "Santa Anna." The plant includes a Babcock & Wilcox boiler, whose connections to the triple expansion engine are shown.

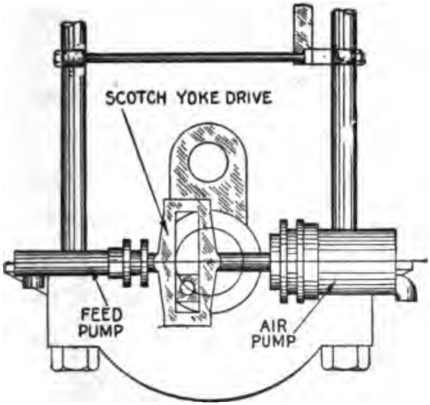


Fig. 2,115.—Scotch yoke drive for pumps. In this arrangement both the air and feed pump plungers are connected rigidly to a Scotch yoke. When arranged as shown the power strokes in both pumps take place at the same time, hence the combined stresses due to both are brought on the Scotch yoke and block. If the pumps were arranged side by side or in tandem on one side of the yoke, the power strokes would alternate and the maximum stress on the yoke would be reduced. The arrangement is compact, but necessarily runs at the same speed as the engine, accordingly in some engines the *r.p.m.* may be too high for satisfactory operation of the pumps.

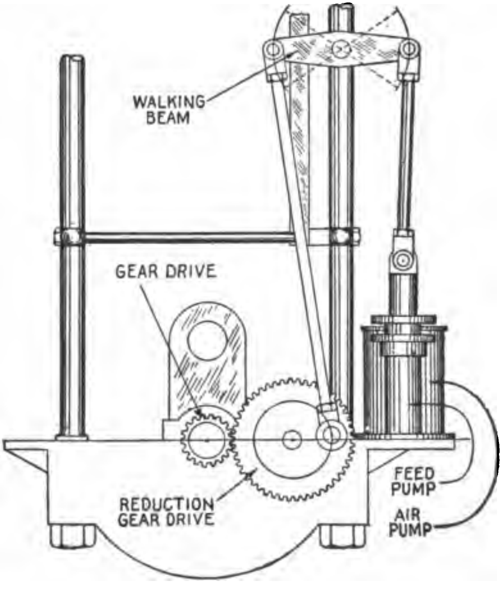
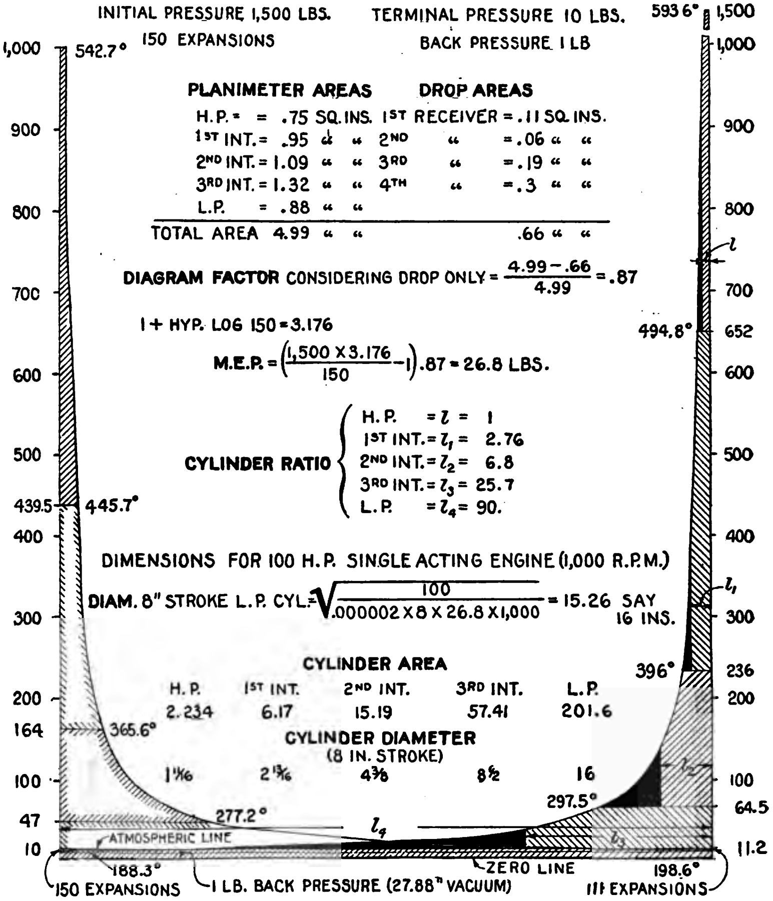


Fig. 2,116.—Geared drive for pumps. For small high speed engines this arrangement permits a speed reduction to any desired number of strokes per minute for the pumps and is a highly satisfactory method of driving pumps on high speed engines.



INITIAL PRESSURE 1,500 LBS. TERMINAL PRESSURE 10 LBS.
150 EXPANSIONS BACK PRESSURE 1 LB

PLANIMETER AREAS		DROP AREAS	
H.P. =	.75 SQ. INS.	1ST RECEIVER =	.11 SQ. INS.
1ST INT. =	.95 " "	2ND " =	.06 " "
2ND INT. =	1.09 " "	3RD " =	.19 " "
3RD INT. =	1.32 " "	4TH " =	.3 " "
L.P. =	.88 " "		
TOTAL AREA	4.99 " "		.66 " "

$$\text{DIAGRAM FACTOR CONSIDERING DROP ONLY} = \frac{4.99-.66}{4.99} = .87$$

$$1 + \text{HYP. LOG } 150 = 3.176$$

$$\text{M.E.P.} = \left(\frac{1{,}500 \times 3.176}{150} - 1\right).87 = 26.8 \text{ LBS.}$$

CYLINDER RATIO

- H.P. = l = 1
- 1ST INT. = l_1 = 2.76
- 2ND INT. = l_2 = 6.8
- 3RD INT. = l_3 = 25.7
- L.P. = l_4 = 90.

DIMENSIONS FOR 100 H.P. SINGLE ACTING ENGINE (1,000 R.P.M.)

$$\text{DIAM. 8" STROKE L.P. CYL.} = \sqrt{\frac{100}{.000002 \times 8 \times 26.8 \times 1{,}000}} = 15.26 \text{ SAY 16 INS.}$$

CYLINDER AREA

H.P.	1ST INT.	2ND INT.	3RD INT.	L.P.
2.234	6.17	15.19	57.41	201.6

CYLINDER DIAMETER (8 IN. STROKE)

H.P.	1ST INT.	2ND INT.	3RD INT.	L.P.
1 11/16	2 13/16	4 3/8	8 1/2	16

FIG. 2,117.—Diagrams illustrating ***quintuple*** or ***five stage*** expansion at initial pressures of 1,000 and 1,500 lbs. absolute. The calculations are for the diagram at the right for 1,500 lbs., 150 expansions, and indicate the cylinder ratio and cylinder dimensions for 100 horse power single acting engine suitable for a high speed boat. In a proposition of this kind the aim should be to obtain equal temperature ranges rather than equal powers for each cylinder, because the main object sought is to reduce the steam consumption to a minimum for the given initial pressure and expansion ratio, in order to reduce the size of the boiler and thus develop the given power with the least weight possible. With proper construction 100° to 150° of superheat could be used, and the net weight per *h.p.* probably further reduced by means of reheating receivers between the *2nd int.* and *l.p.* cylinders.

CHAPTER 39

INSTALLATION

Location.—As a rule, an engine should be located as centrally as possible with respect to the distribution of power, that is, in case a long line of shafting is to be driven, the best transmission is obtained by placing the engine at the middle of the

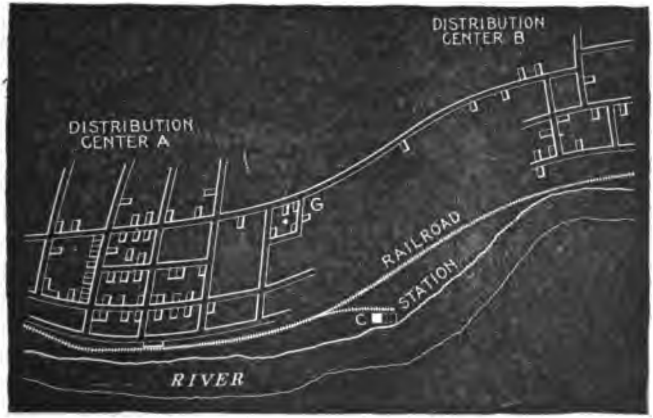


Fig. 2,118.—Station location. The figure shows two distribution centers as a town A, and suburb B, supplied with electricity from one station. For minimum cost of copper the location of the station would be at G, the center of gravity. However, it is very rarely that this is the best location. For instance at C, land is cheaper than at G, and there is room for future extension to the station, as shown by the dotted lines, whereas at G, only enough land is available for present requirements. Moreover C, is near the railroad where coal may be obtained without the expense of cartage, and being located at the river, the plant may be run condensing thus effecting considerable economy. The conditions may sometimes be such that any one of the advantages to be secured by locating the station at C, may more than offset the additional cost of copper.

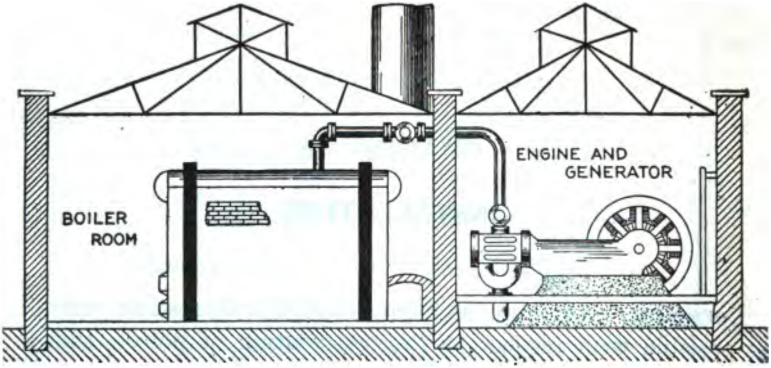


Fig. 2,119.—Location of engine and boiler. Ideal conditions for steam economy require that the steam pipe should be as short and direct as possible, while for minimum wear, the engine should be in a separate room so its bearings and working parts will be protected from the ashes, coal, dust, etc., of the boiler room.

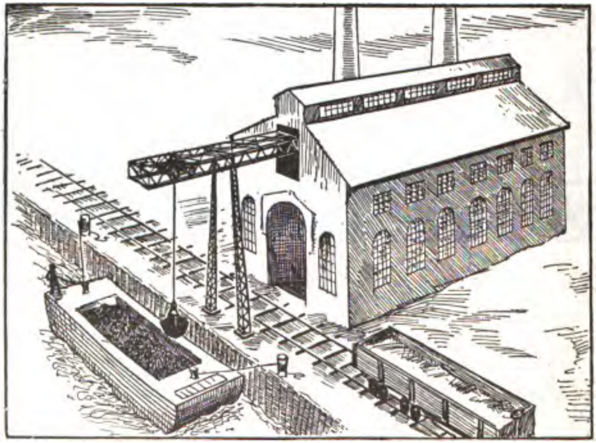


Fig. 2,120.—View of a station admirably located with respect to transportation of the coal supply. As shown, the coal may be obtained either by boat or rail, and with modern machinery for conveying the coal to the interior of the station, the transportation coast is reduced to a minimum.

line. When thus located, more power can be transmitted with a given size shaft, than when placed at the end.

With respect to steam consumption, the nearer the engine is to the boiler, the greater the efficiency. Long lines of steam pipe, though protected by the best covering, will cause more or less condensation. The engine should be placed in a separate room from the boiler, so that the working parts will be free from ashes and coal dust, the presence of which greatly increases wear.

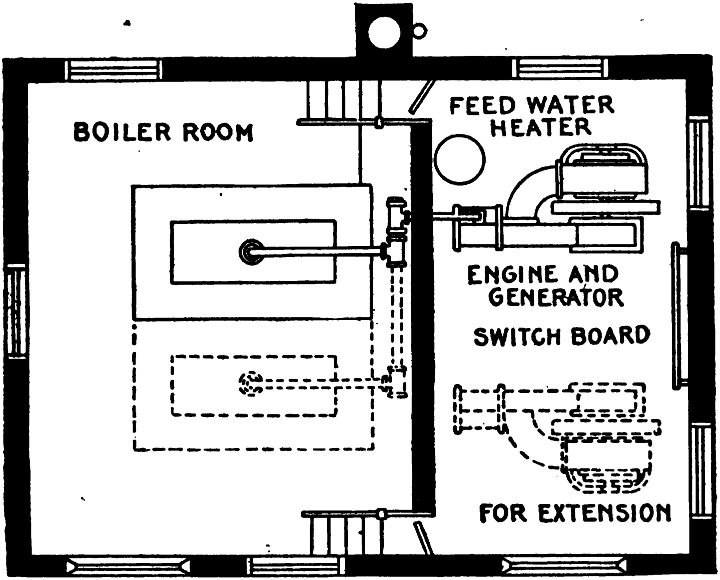


FIG. 2,121.—Plan of station arranged for extension. The space required for a central station depends upon the number and kind of lights to be supplied, and upon the character and arrangement of the machinery. In calculating the size of building required, two things must be carefully considered: first, the building must be adapted to the plant to be installed in the beginning; and second, it must be arranged so that enlargement can be made without disarranging or interfering with the plant already in existence. This is usually best secured by providing for expansion in one or two definite directions, the building being made large enough to accommodate additional units that will be necessary at some future time because of the growth of the community and consequent increased demand for electric current.

In a new plant there should be little difficulty in determining the correct location, but in case of a growing plant, where larger engines are to be installed to replace those that have become too small for the increased power demands, the problem is more complex, and requires good judgment. In some cases the loss of time due to shutting down a plant to remove an old engine and foundation, build new foundations and erect a new engine upon the site of the old one, will more than offset the gain by having a compact installation.

There are many points to be considered in deciding upon a location, but each case has its peculiarities.

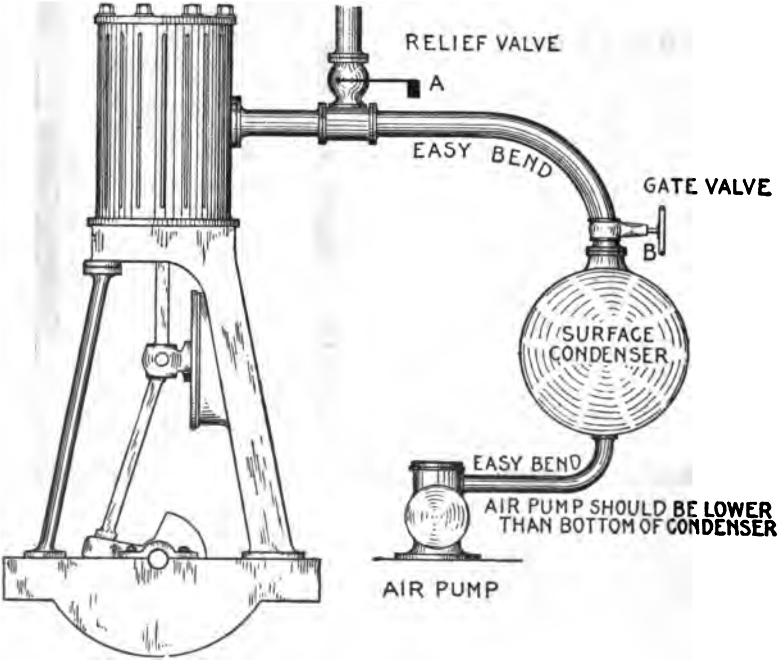


FIG. 2,122.—View of engine and condenser, showing how to arrange the piping to secure good vacuum. *Locate the condenser as near the engine as possible*; **use easy bends** *instead of elbows*; *place the pump* **below** *bottom of condenser so the water will drain to pump.* At A, is a relief valve, for protection in case the condenser become flooded through failure of the pump, and at B, is a gate valve to shut off condenser in case atmospheric exhaust is desired to permit repairs to be made to condenser during operation. **A water seal** should be maintained on the relief valve and **special attention** *should be given to the stuffing box* of the gate valve **to prevent air leakage.** *The discharge valve of the pump should be water sealed.*

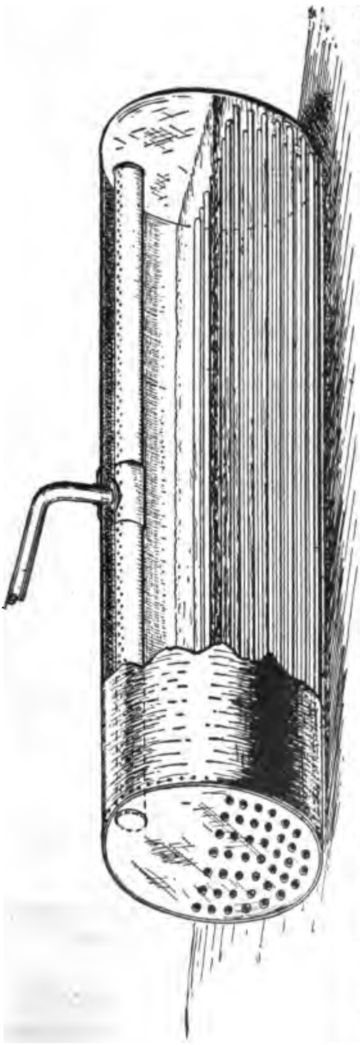


FIG. 2,123.—So called *"dry pipe"* for horizontal boiler; it is connected to the main outlet and its upper surface is perforated with small holes, the far end being closed. With this arrangement steam is taken from the boiler over a large area, so that it will contain very little moisture. *All horizontal boilers without a dome should be fitted with a dry pipe*; most engineers do not realize the importance of obtaining dry steam for engine operation.

In locating the engine due regard should be had with respect to the pipe connections, so that all joints will be easily accessible for packing and repair.

Foundations.—It is essential that a steam engine be placed upon a solid unyielding foundation. Only the best material should be used and in liberal quantities. Engine builders furnish blue prints giving the proper dimensions for the foundation, assuming that the bottom rests on solid ground. If this be not reached at the depth indicated, the excavation should be carried deeper until firm soil is reached.

Concrete makes a most excellent foundation and is largely used instead of brick work. A good mixture for this purpose consists of, one part Portland cement, two parts clean sharp sand, and three parts broken stone. *Only the best grade of cement should be used.*

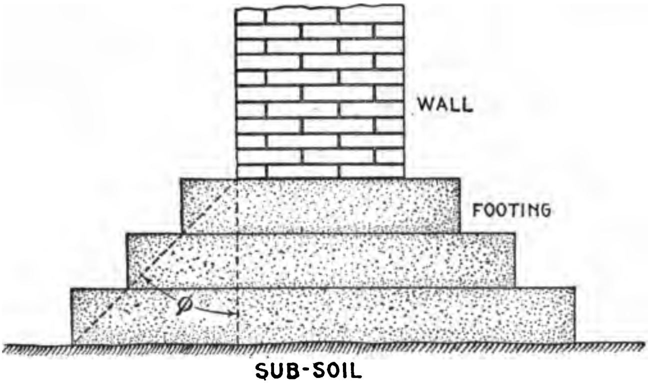


Fig. 2,124.—Angle for foundation **footing.** ***In ordinary practice*** the footing courses upon which the walls of the building proper rest, consist of blocks or slabs of stone as large as are available and convenient to handle. Footings of brick or concrete are also used in very soft soils; footings consisting of timber grillage are often employed. A grillage of iron or steel beams has also been used successfully. The inclination of the angle ϕ, of footing should be about as follows: for metal footings 75°; for stone, 60°; for concrete, 45°; for brick, 30°. Damp proof courses of slate, or layer of asphalt are laid in or on the foundations or lower walls to prevent moisture arising or penetrating by capillary attraction.

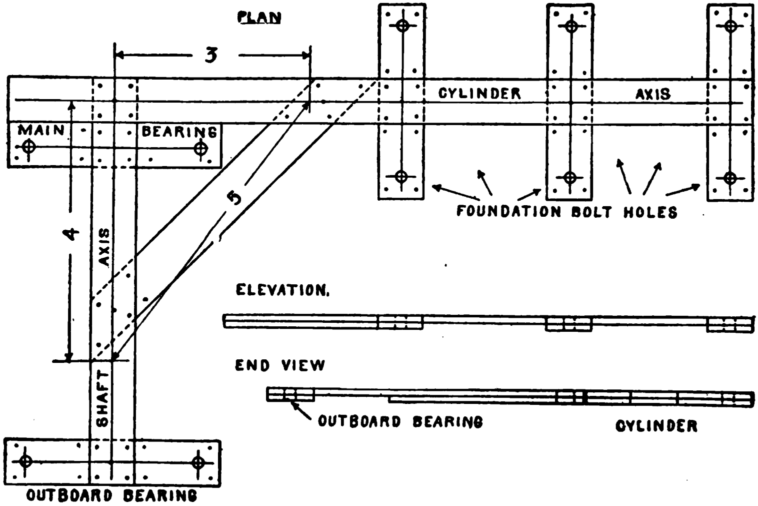


Fig. 2,125.—Template for engine foundation.

When the excavation has been carried down to the required depth, its surface should be levelled and thoroughly tamped, keeping it quite damp while the tamping is being done.

A wooden template should be made with holes corresponding to those in the engine base as shown in fig. 2,125. When this is in place, as in fig. 2,126, tin pipes, at least two inches larger in diameter than the bolts, are suspended centrally from each hole, reaching to the anchor space, so that when the concrete is poured, there will be a margin of space around each bolt permitting lateral adjustment to allow for any minute errors in measurements, and to facilitate the removal of a bolt in case of breakage. After these pipes and the necessary forms are in place, the concrete should be prepared in sufficient quantity that the volume of the foundation may be filled with one pouring. The foundation should be completed at least fifteen days before the engine is placed upon it, by which time it should have become sufficiently hard to resist the weight of the engine.

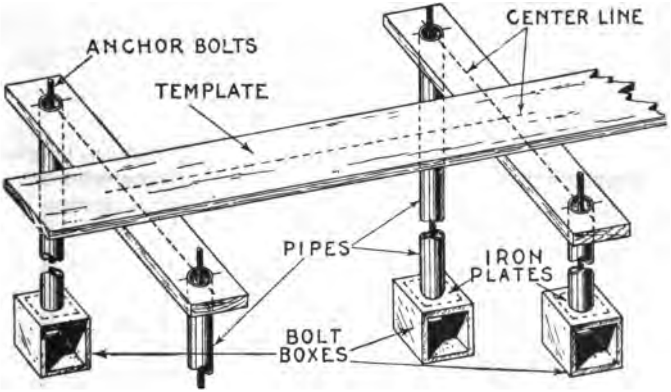


FIG. 2,126.—View showing part of template for locating anchor bolt centers, pipes through which the bolts pass and bolt boxes at lower end of bolts. The completed foundation is shown in fig. 2,127, with template removed. The template is made of plain boards upon which the center lines are drawn, and bolt center located. Holes are bored at the bolt centers to permit insertion of the pipes as shown.

Placing the Main Castings.—When the foundation is complete, and the cap stones in position, the top nuts may be removed from the foundation bolts, and the latter dropped down in the pockets out of the way. The main parts should be brought into the engine room in such order that those belonging furthest from the entrance will come first, thus avoiding any unnecessary movement of these parts.

The half section of the fly wheel which does not contain the keyway is placed in the wheel pit. Twenty-four iron wedges, say two by six inches, tapering from seven-eighths to quite sharp will now be needed.

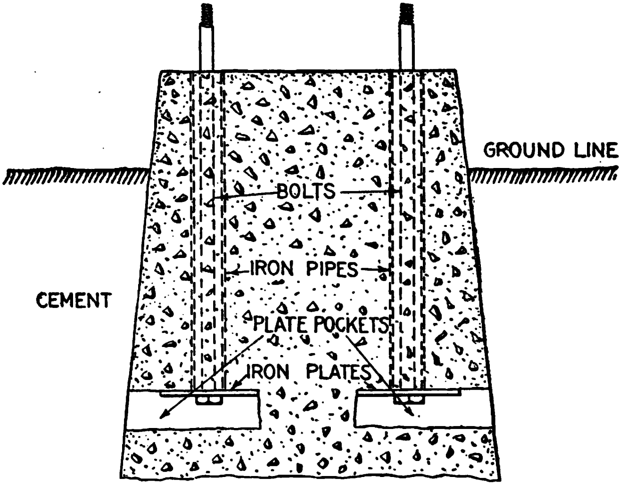
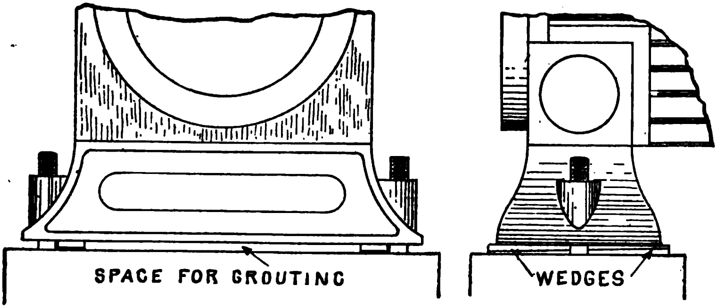


Fig. 2,127.—Concrete foundation, showing method of installing the anchor bolts.

After removing the piston, piston rod, cross head and connecting rod, the cylinder, frame, and outboard bearing are placed in position with the wedges well entered under the feet. The proper method of placing the wedges is shown in figs. 2,128 and 2,129. The cylinder and frame should be bolted together, being careful to remove any foreign matter from the surface of the joint. In bolting them together the nuts should be tightened evenly by taking up on opposite bolts so as not to throw the parts out of line.

The foundation bolts should be raised and the top nuts put



Figs. 2,128 and 2,129.—End and side views of one end of cylinder shows placement of wedges and space for grouting.

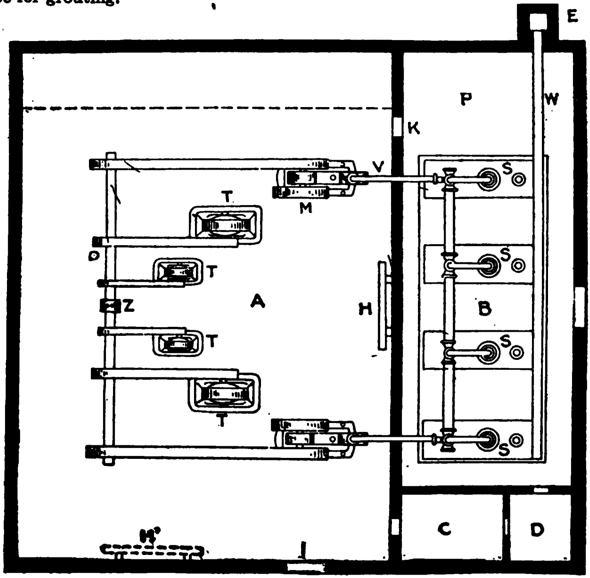
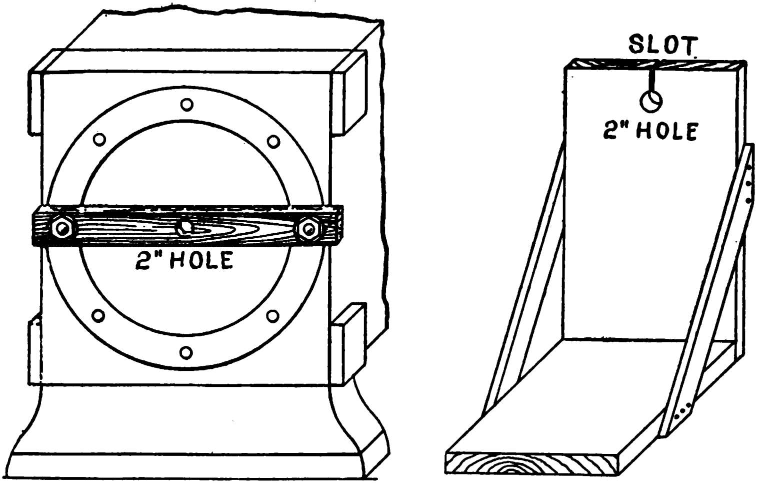


Fig. 2,130.—Floor plan of an electrical station having a belted drive with counter shaft.

on loosely, leaving ample room for levelling. Care should be taken to see that none of the bolts will bind when the engine is shifted.



FIGS. 2,131 and 2,132.—Template for locating cylinder axis and standard.

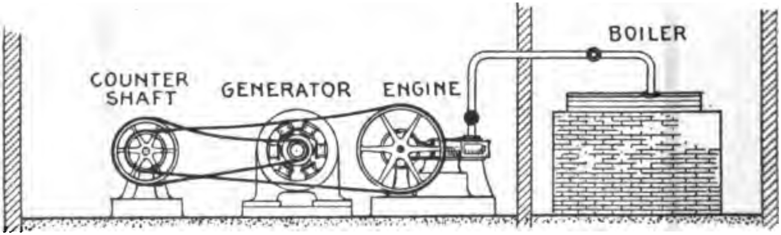


FIG. 2,133.—Elevation of station having a belted drive with countershaft, as shown in plan in fig. 2,130.

Alignment.—A board should be fastened across the flange of the cylinder on two opposite studs, as shown in fig. 2,131, the approximate center located, and a one-inch hole bored at this point. A standard must be made, as shown in fig. 2,132, long enough to project from the floor a few inches higher than the center line of the engine; this should be securely nailed to the floor, being placed so that the hole will be approximately in line with the center line of the engine. Having procured some fine piano wire, a small stick is fastened to one end of the wire, and drawn through the stuffing box and the hole in the board attached to the cylinder flange. The other end is fastened to a small bent lever made of three-eighths inch round iron as shown in fig. 2,135,

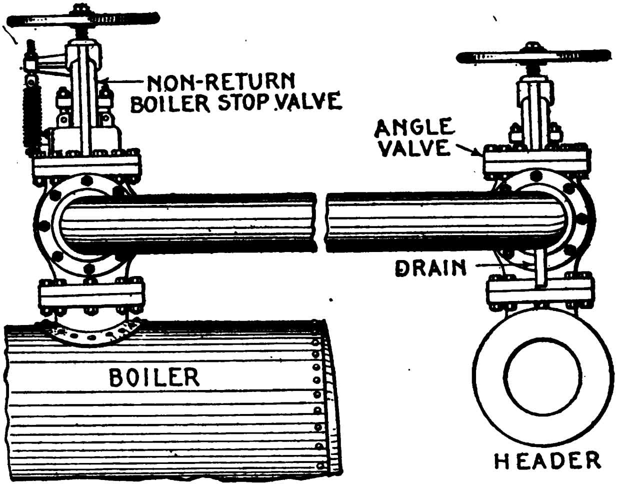


FIG. 2,134.—Method of connecting a header to a battery of boilers. Where two or more boilers are connected to a single header, the use of a reliable non-return boiler stop valve is necessary, and in some countries their installation is compulsory. A non-return boiler stop valve will instantly close should the pressure in the boiler to which it is attached suddenly decrease below that in the header, and thereby prevent the entrance of steam from the other boilers of the battery. This sudden decrease in pressure may be caused by a ruptured fitting or the blowing out of a tube, in which event an ordinary stop valve taking the place of a non-return boiler stop valve would be inadequate, as the loss of steam from the other boilers of the battery would be tremendous before an ordinary valve could be reached and closed, assuming that it would be possible to do so, which in the majority of cases it would not. Should it be desired to cut out a boiler for cleaning or repairs, the non-return boiler stop valve will not permit steam to enter the boiler from the header, even should the hand-wheel be operated for this purpose, as it cannot be opened by hand, but can, however, be closed. A non-return boiler stop valve should be attached to each boiler and connected to an angle valve on the header. A pipe bend should be used for connecting the valves, as this will allow for expansion and contraction. The pipe should slope a trifle downward toward the header and a suitable drain provided. This drain should be opened and all water permitted to escape before the angle valve is opened, thereby preventing any damage due to water hammer.

and passed through the slot to the hole in the standard, as in fig. 2,136. By letting the rod rest against the back of the standard and turning the crank, the line may be drawn up with sufficient tension so there will be practically no sag. The line is secured

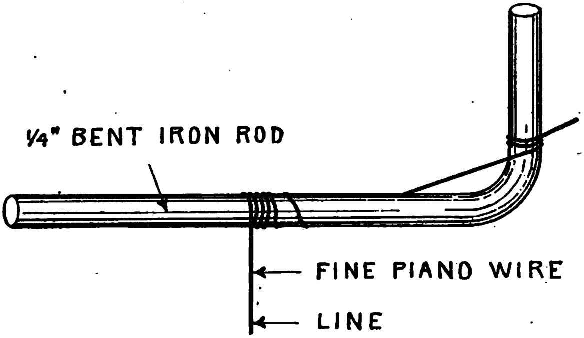


Fig. 2,135.—Small bent lever of ⅜ round iron to secure wire center line taut in position.

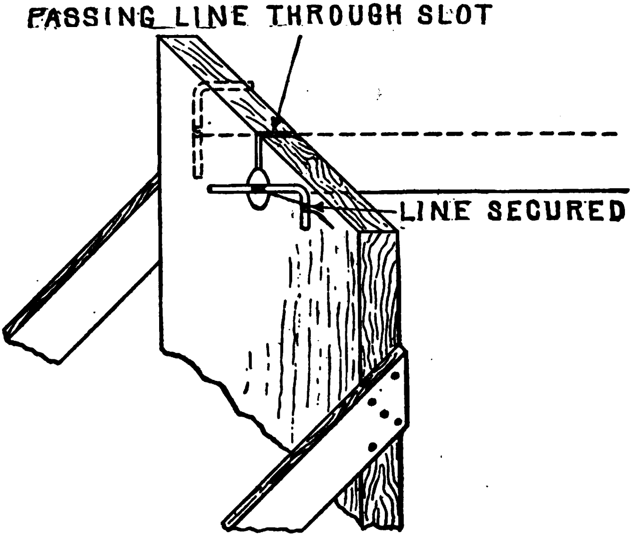
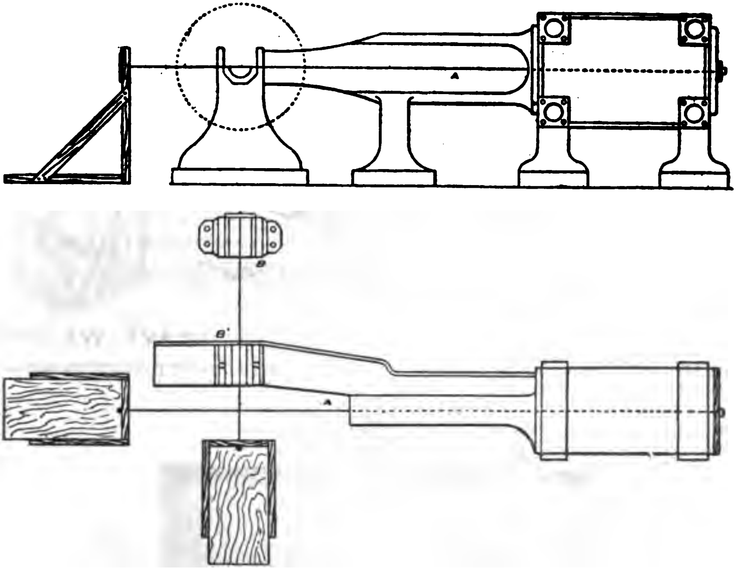


Fig. 2,136.—Method of securing wire center line.

by pushing the crank around till it engages with the back of the standard, as in fig. 2,136, the position of the line with respect to the engine being shown in figs. 2,137 and 2,138.

Now with a pair of calipers, set to a length equal to one-half the diameter of the counter-bore, the piano wire is centered at the head end of the cylinder. The wire is easily shifted to the desired position by moving the small stick to which it is fastened, the hole in the cross piece being sufficiently large to permit this adjustment. Having centered the wire at the head end, it is next centered with the stuffing box, by shifting its position on the standard. The wire now represents the center line of the engine, with respect to which the adjustment of all the other parts must agree.



Figs. 2,137 and 2,138.—Elevation and plan of an engine with outboard bearing showing cylinder, and shaft center line alignment wires in position.

The first step after locating the center line is to adjust the position of the outboard bearing so that the shaft will be at right angles with the cylinder. To do this a second line of piano wire is run through the outboard, and main bearing to a standard opposite the latter. This line is centered and adjusted at right angles to the center line as shown in fig. 2,138; by shifting the outboard bearing, its correct position being determined by use of a square.

The shaft may now be placed in position and the adjustment tested by the method of figs. 2,141 to 2,144.

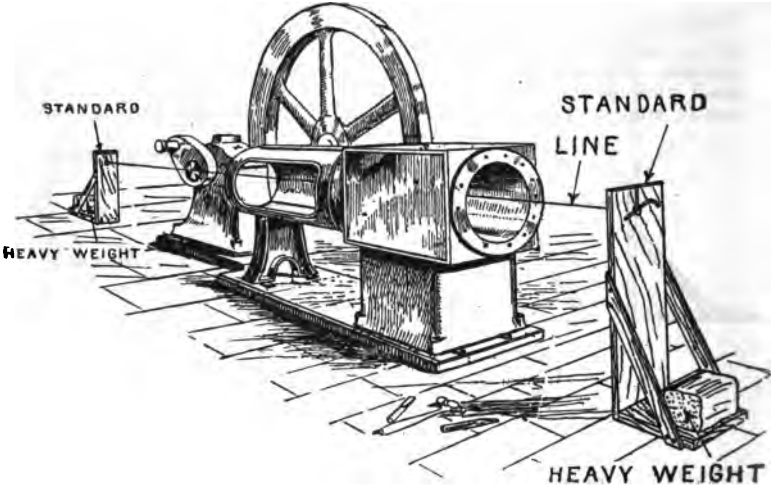


FIG. 2,139.—View of engine showing center line alignment wire secured in position by a standard at each end of the engine.

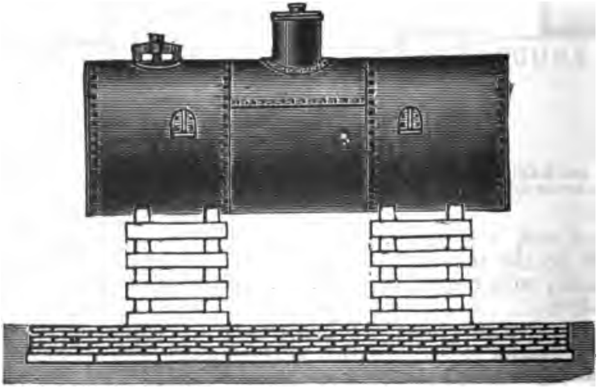
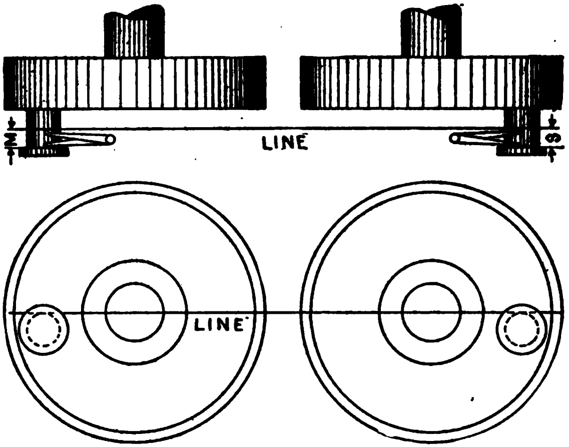
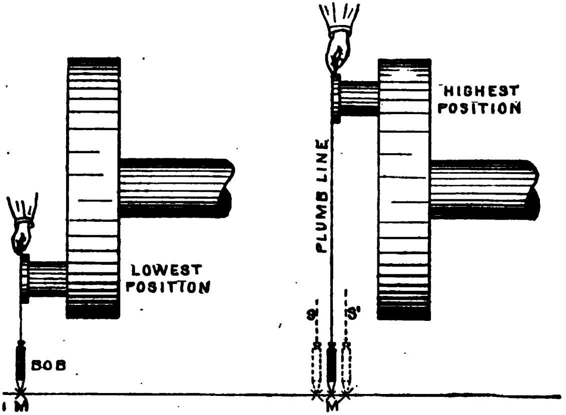


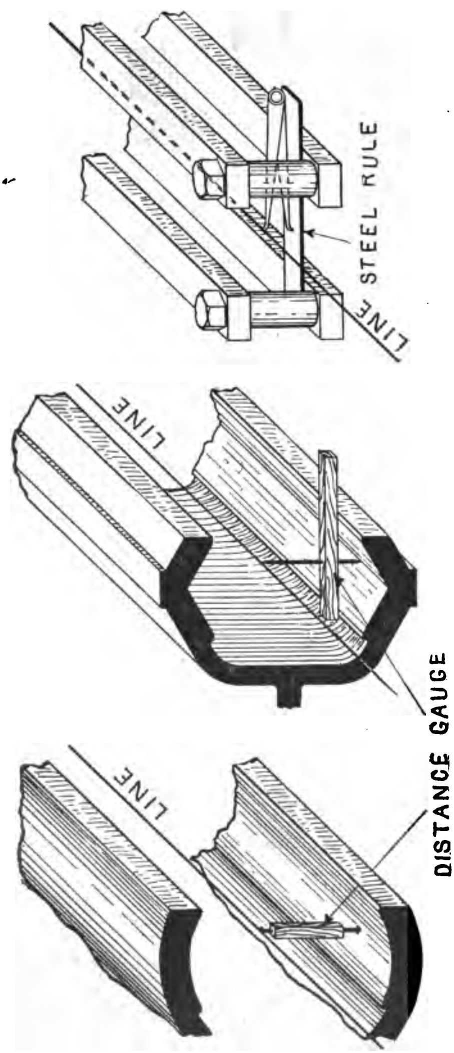
FIG. 2,140.—Method of supporting a horizontal return tubular boiler while erecting.



FIGS. 2,141 and 2,142.—Aligning shaft with respect to the cylinder axis. The crank pin is rotated until it comes in contact with the alignment wire, first at one end of the stroke, and then at the other; in each case the distance from the wire to the crank pin flange M and S, is measured with inside calipers, and the shaft adjusted until M=S, when the shaft will be at right angles with the cylinder axis.



FIGS. 2,143 and 2,144.—Method of testing horizontal adjustment of the shaft after levelling with a spirit level. As shown, the shaft is turned until the crank pin comes first in the highest, and then in the lowest position, and plumb bob being applied in each position. If the point of the bob touches the same point M, for each position, then the shaft is horizontal; if it fall at points such as S and S′, for the two positions, the shaft is not level and must be further adjusted until the bob touches the same point in each position.



FIGS. 2,145 to 2,147.—Various methods of testing the alignment of the guides adapted to the different types of guide shown.

A spirit level is used for the horizontal adjustment of the shaft, and the work tested with a plumb bob as shown in figs. 2,143 and 2,144.

The guides will probably need no adjustment, but it is well to test the truth of their alignment. Various methods are used, depending on the type of guides as shown in figs. 2,145 to 2,147.

Grouting.—The spaces between the cap stones and feet of the engine left clean by the insertion of the wedges must be filled with some suitable material, giving a firm bearing. The operation of filling these spaces is called "grouting."

When satisfied that the engine is level and in line, attention should be given to the foundation bolts to see that all have an equal bearing, after which the nuts should be firmly screwed down. The lines and level should again be tried in all directions, because it is possible to spring the engine down, or to one side in tightening the bolts; this must be remedied by the wedges, and another trial made.*

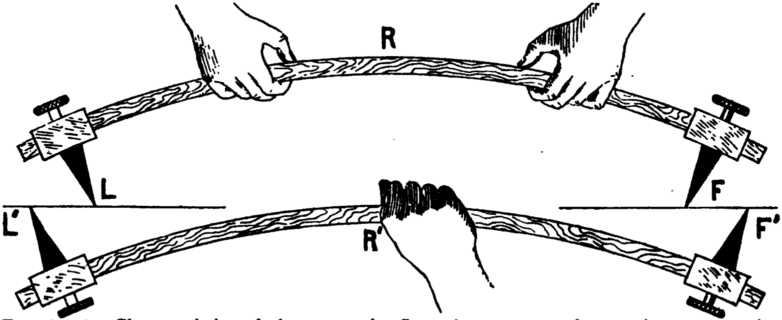
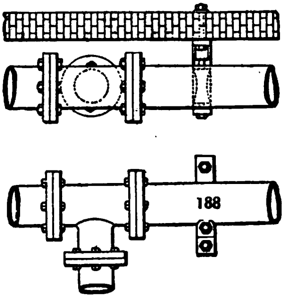


FIG. 2,148.—Characteristics of the trammel. *In using* a trammel, care is necessary in holding the bar to avoid serious mistakes. Since a trammel is frequently used for transferring measurements of large dimensions, the above caution should be noted. If a pair of ordinary trammels be put upon an ordinary wooden bar, and rest with their points upon a surface, the points can be brought nearer together as positions L, F, by bowing the bar backward in the center R, of its length, and also be made to move outward as positions L'F', by bowing the bar forward at its center R', the two cases being represented in an exaggerated degree in the figure. Hence a very stiff bar should be used and handled carefully to prevent any distortion.



FIGS. 2,149 and 2,150.—Method of preventing vibration and of supporting pipes. The figures show top and side views of a main header carried in suitable frames fitted with adjustable roller. While the pipe is illustrated as resting on the adjustable rollers, nevertheless the rollers may also be placed at the sides or on top of the pipe to prevent vibration, or in cases where the thrust from a horizontal or vertical branch has to be provided for. This arrangement will take care of the vibration without in any way preventing the free expansion and contraction of the pipe.

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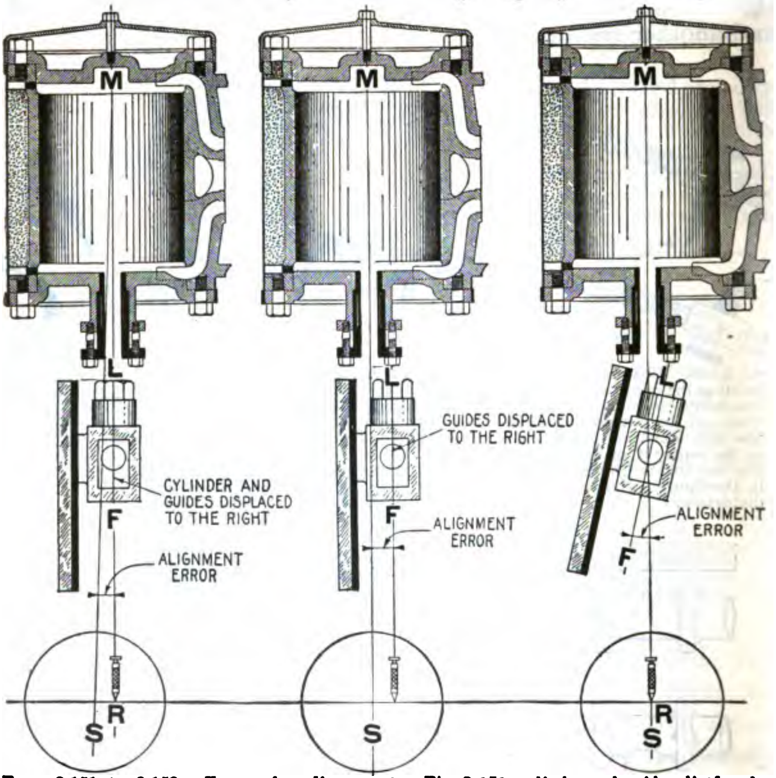
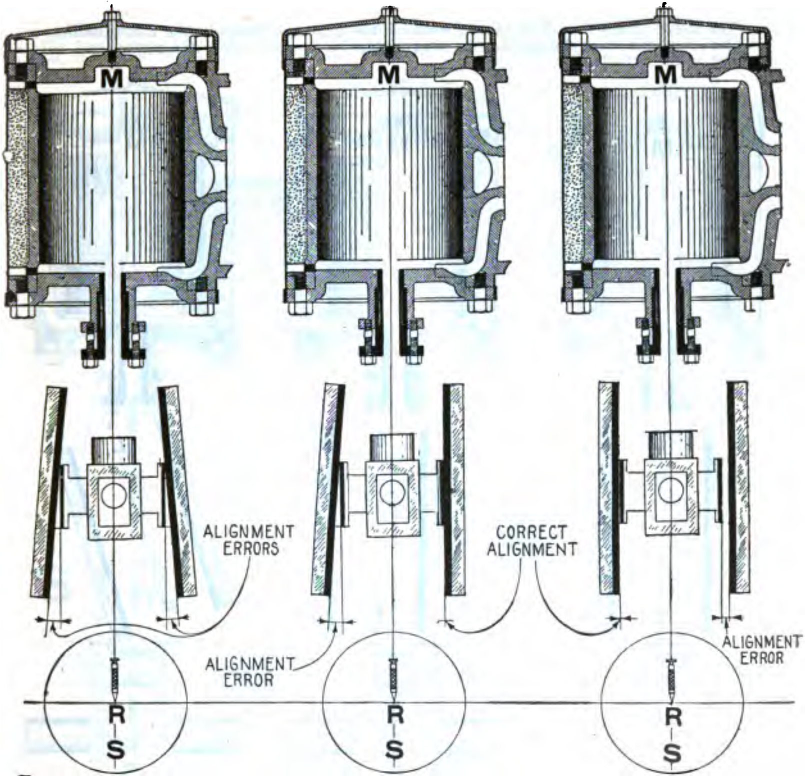
*NOTE.—When the bolts may be tightened quite firmly without disturbing the alignment or level, preparation may be made for grouting. Some waste should be wrapped around the foundation bolts, and stuffed down an inch or so in the bolt holes to prevent the entrance of the filling which would grout the bolts. There are several materials used for grouting of which the following are recommended. If the spaces between the cap stones and castings be three-eighths inch or less, a filling composed of one part antimony and seven parts lead will make a satisfactory joint. If there be very thin spaces to fill, kerosene should be sprayed into the opening and the hot metal quickly poured. For an opening one-half inch or larger, the best grade of Portland Cement should be used, mixed clear and quite thin. After the filling has been given sufficient time to thoroughly harden (24 hours for cement), all the foundation bolts should be permanently tightened.

To retain the filling in place before it hardens, a dam of sand should be made all around each foot, with sufficient opening to pour the filling.


M
L
CYLINDER AND GUIDES DISPLACED TO THE RIGHT
F
ALIGNMENT ERROR
S
R
M
L
GUIDES DISPLACED TO THE RIGHT
F
ALIGNMENT ERROR
S
M
L
ALIGNMENT ERROR
F
R
S

Figs. 2,151 to 2,153.—*Errors in alignment.* Fig. 2,151, ***cylinder and guides displaced to right of vertical axis.*** Obviously, the cylinder and guides should be shifted to the ***left.*** However, as the engine will run satisfactorily as it is, realignment would hardly be necessary, especially if the expense incurred be great. Fig. 2,152, ***cylinder axis in line with shaft center, but guides are displaced to right.*** If correcting the error in alignment of guides be expensive, they could be left as they are by adjusting the cross head gibs so as to compensate for the error. Fig. 2,153, ***axis of guides not parallel with cylinder axis.*** Engine will not work satisfactorily with this error if serious. **MS**, line through center of cylinder end and center of shaft; **MR**, plumb line through center of cylinder end; **LF**, guide axis.

Assembling the Reciprocating Parts.—The shaft journal should be tested, and scraped to a good bearing before placing the fly wheel.

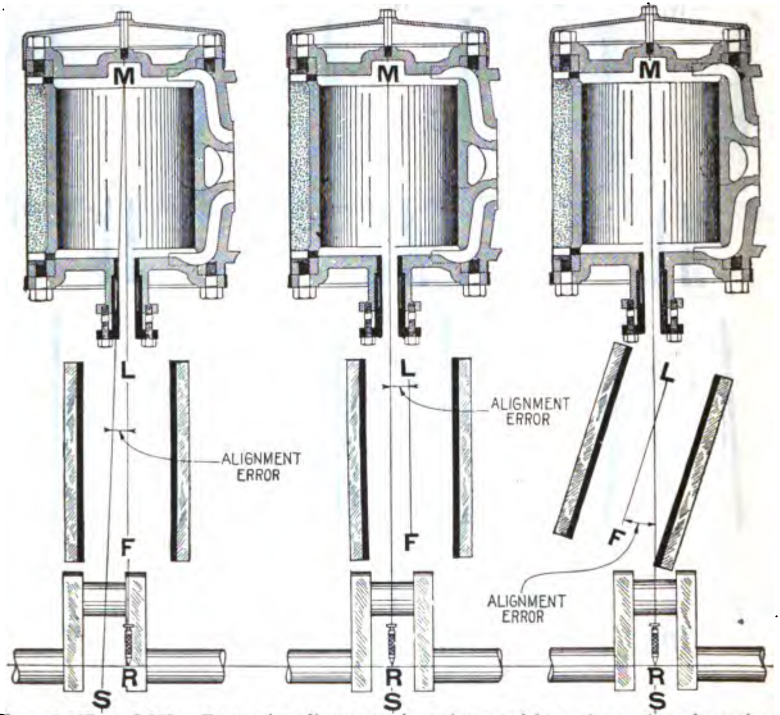


FIGS. 2,154 to 2,156.—***Errors in alignment.*** Fig. 2,154, *cross head guides not parallel.* Since only one gib of the cross head bears on the guide at a time, the tendency of the error is to cause the piston to bind at the cylinder ends, or if much out, to cause the cross head to bind at one end. Fig. 2,155, *one guide not parallel to cylinder axis.* Where adjustable flat guides are used as on some marine engines, an error in parallelism of one guide may be sometimes detected by faulty running in forward or reverse motion according to which guide be out of alignment. Fig. 2,156, *guides at different distances from cross head axis.* If the far guide be opposite the working gib it will cause *slapping*, that is a knock during admission. **MS**, cylinder axis; **MR**, plumb line through center of cylinder end.

With the shaft turned so that the key seat is uppermost, the key should be tried in both the shaft, and the fly wheel hub. In case of a split wheel, the key should not be driven in place, but the wheel clamped with the key in piston.

***NOTE.—With proper tackle, the shaft must be hoisted from the bearings, the eccentric put in position, and the surface of the bearings given a light coat of red lead. If, on replacing the shaft, and turning it a few times, there be any "high spots," the red lead will be rubbed off these points, thus indicating that the surface at these places is too high, hence it should be scraped, and the operation repeated until all the bearing surface is true with the shaft.**

The steam fitters should be put to work on the piping, as soon as the main castings are in place, so that the cylinder may be thoroughly blown out before the piston and valves are put in the cylinder.*



FIGS. 2,157 to 2,159.—***Errors in alignment,*** in a plane at right angles to that of rotation of the crank. Fig. 2,157, ***plumb line does not pass through center of crank pin.*** This error generally arises when two or more cylinders are bolted together and the dimension between their centers differs from that between their respective crank pins. Fig. 2,158, ***cylinder and guide axis, parallel but do not coincide.*** Fig. 2,159, ***cylinder and guide axis not parallel*** with adjustable guides of the type shown. These errors are easily corrected. The plumb line **MR**, falls to the right of **MS**, in fig. 2,157, and coincides with it in figs. 2,158 and 2,159. **LF**, is the axis of the guides.

If the piston rod be attached to the cross head with a key no clearance adjustment is necessary, but with a threaded joint the proper length of rod must be determined by locating the "striking points."

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*NOTE.—Blowing out the cylinder will remove any accumulation of grit, etc. If the engine is to be run condensing this operation should be performed before connecting the exhaust pipe to the condenser. While the pipe fitting is in progress, the valve gear can be assembled, connecting rod and cross head, leaving the piston and valves for the last. When the cylinder has been thoroughly blown out with steam, the head may be removed and the interior wiped clean before inserting the piston.

With the engine on either center, the rod is turned till the piston comes in contact with the cylinder head, and the number of threads exposed on the rod counted; the operation is repeated for the other center, thus determining the number of threads corresponding to both clearances. Taking one-half the difference, and turning the rod to correspond with this point, the clearances are thus equalized.

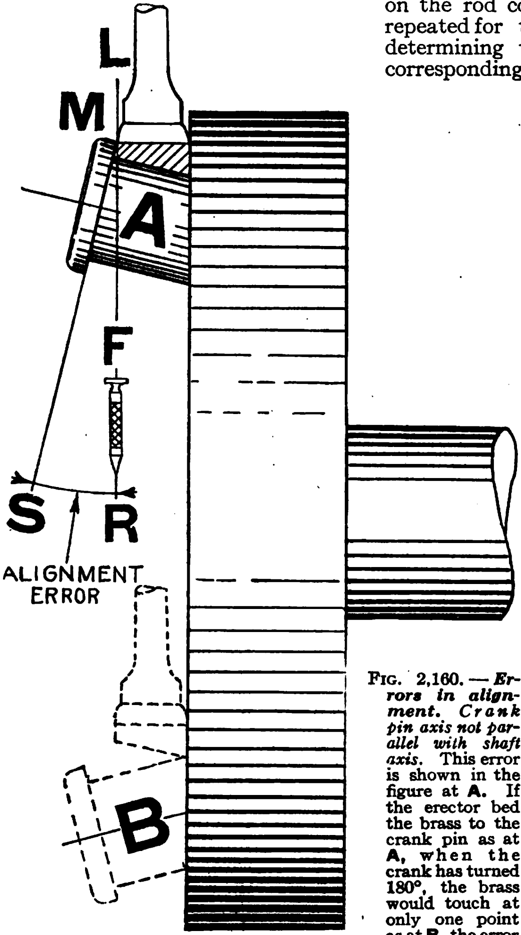


FIG. 2,160.—*Errors in alignment. Crank pin axis not parallel with shaft axis.* This error is shown in the figure at **A**. If the erector bed the brass to the crank pin as at **A**, when the crank has turned 180°, the brass would touch at only one point as at **B**, the error at this point being doubled. The angle between axis **MS**, and **LFR**, shows crank pin error. The running results depend on the design of the rod end, and conditions under which the brasses are held.

The valves should now be placed in position and set as instructed in the chapters on valve setting, noting carefully the precautions to be taken, especially with the Corliss valve gear.

Marine Engine Shaft Alignment.—When installing a marine engine in a vessel, the crank and propeller shafts should be aligned with much care, as poor alignment of these parts causes considerable friction, wear and loss of power.

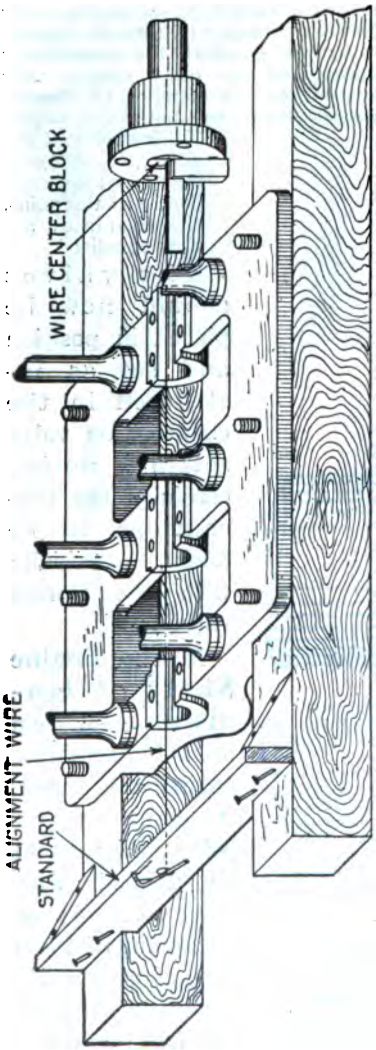


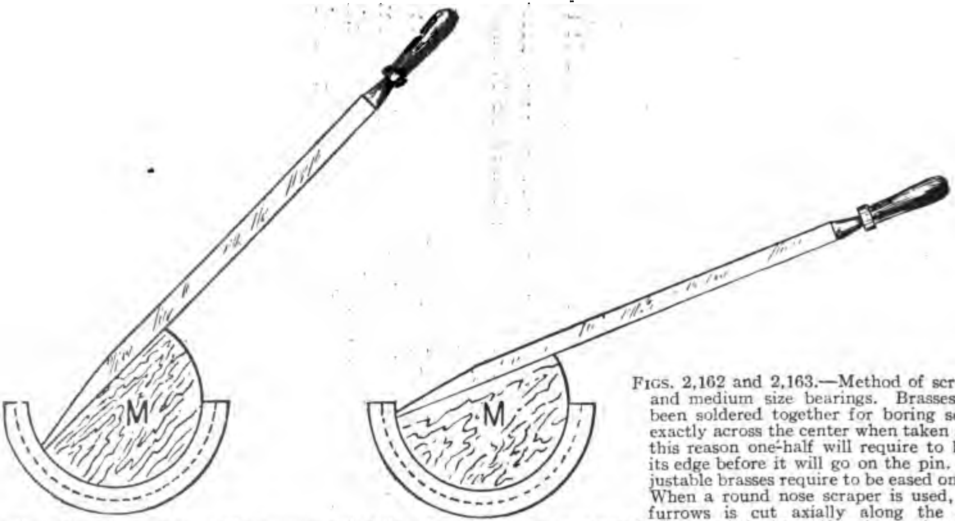
FIG. 2,161.—Method of aligning the bedplate of a marine engine with the propeller shaft. In the case of a new vessel the engine may be approximately aligned before launching. This should be done before the propeller shaft is put in position. The alignment wire is stretched from the standard in front of the bed plate through the stern tube. The final adjustments should be made after the vessel is in the water, in the manner shown above, because especially in the case of light boats, the hull changes its shape more or less with the redistribution of supporting pressure.

First, the caps of the main bearings should be taken off and the crank shaft removed to make room for the center line.

On all direct coupled engines the male side of the coupling should be attached to the crank shaft, leaving the female side on the propeller shaft which is more conveniently centered.

A center line of fine piano wire must be stretched from the center of the female coupling through the main bearings to a standard as shown in fig. 2,161.

In fastening this line to the coupling, a small piece of wood should be cut to size and forced into the hole in the coupling, leaving a space be tween it and the end of the shaft. The center of the shaft is located on this piece of wood and a small hole the size of the wire bored. One end of the wire is passed through the hole and fastened; the other end is carried through the bearings to the standard, and drawn taut.



FIGS. 2,162 and 2,163.—Method of scraping small and medium size bearings. Brasses that have been soldered together for boring seldom split exactly across the center when taken apart. For this reason one-half will require to be eased at its edge before it will go on the pin. Many adjustable brasses require to be eased on their sides. When a round nose scraper is used, a series of furrows is cut axially along the brass; and further, these furrows cannot be commenced at the edge of the brass, nor can they be carried with uniformity to the other end of the brass. When a half-round scraper is used the results are far from perfect. In the first place, a half-round scraper is not a tool on which a great deal of pressure can be easily applied, owing to the manner in which it has to be handled. A further disadvantage lies in the fact that it is practically impossible to apply the operating force at a constant angle to the surface being scraped. Exactly the same remarks apply to the cutting angle, which cannot be maintained at a constant angle, to the cylindrical bore of the brass. The illustration shows a very useful method of scraping out small and medium sized brasses. In the figure, M, is an approximately semi-cylindrical piece of wood which is placed in the brass, the latter being securely held in the vise or elsewhere. The flat scraper as shown, rests with its flat side upon this wooden guide, and its cutting edge against the surface of the brass. Now, as the scraper handle is lowered, the wooden rest revolves in the brass, and as the scraper rests with its flat side upon M, the flat side always remains constant. It will be obvious to any practical engineer that a great deal of pressure can in this way be applied to the cutting edge, and long, wide uniform cuts can be taken with ease.

The wire is now set in line with the propeller shaft by shifting its position at the standard.

In order to true the line, a square is placed on the face ot the coupling allowing the blade to extend over the line as shown in the figure; the square should be tried on top and bottom, and on each side moving the end of the wire attached to the standard till the line comes true with the square in all these positions.

With small inside calipers measurements are to be taken from the sides of the journals to determine if the line be central with the bearings. If not, the bed plate must be moved sidewise till the bearings are in line with the wire.

A similar adjustment must be made in a vertical direction.

The calipers are set to the half-diameter of the shaft, making the setting scant to allow for the half diameter of the wire, and measurements taken from the bottom of the journals. The bed plate is adjusted vertically till the calipers register with the line, then the bearings may be said to be in line with the propeller shaft.

In making the vertical adjustment, suitable shims should be provided for insertion between the bed plate and keelsons; these should be made of thin sheet metal.

When the adjustments have been made, the nuts on the lag screws or foundation bolts should be firmly set up; this however, will probably put the bearings out of line vertically, in which case additional shims will be required to bring the bearings back in line.

CHAPTER 40

LUBRICANTS

Few owners and managers of plants realize the necessity of proper lubricants. It is in the selection of a lubricant as well as in the application that care and skill are required. For instance, a lubricant which would make a large shaft run smooth and cool in its bearings might be quite unsatisfactory if applied in some other place as for example, a light, high speed spindle.

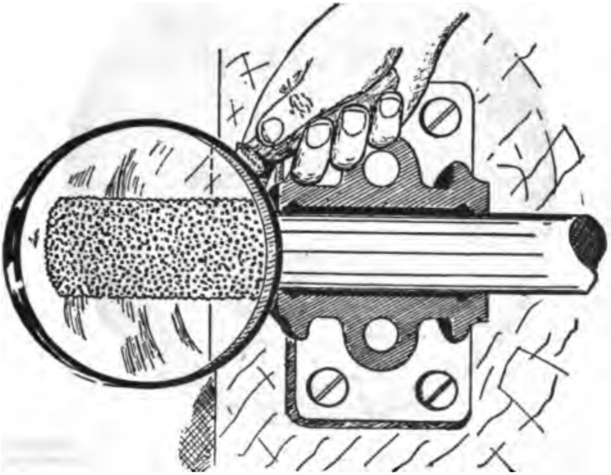


FIG. 2,164.—Magnified view of a shaft showing its rough granular structure. ***In operation,*** these minute irregularities interlock and act as a retarding force, or ***frictional resistance.*** Hence, the necessity for lubrication which prevents actual contact by presenting a thin intervening film against which the surfaces rub. The magnifying glass shown above is simply suggestive of magnification, in fact, to see the rough granular structure the shaft would have to be viewed under a microscope.

The duty of a lubricant is to reduce friction; the lubricant accomplishes this by keeping the parts separate, being pressed out into a thin film on which the moving part rubs thus preventing direct contact. The necessity for this is apparent from fig. 2,164, which is an imaginative view of a shaft showing its rough granular structure.

Metal surfaces may appear smooth to the eye, and feel smooth to the touch, but when examined under the microscope, they will show minute irregularities, ***and appear something like nutmeg graters.***

FIG. 2,165.—***Fluidity or viscosity test.*** Viscosity represents the ***flowing quality*** of an oil and is determined by noting the number of seconds taken to pass a certain quantity of the oil at a specified temperature through a standard size orifice made for the purpose.

When two metal surfaces are brought in contact, these minute irregularities interlock and act as a retarding force. That is why it is impossible to run machinery without lubrication of some kind.

It is evident that lubricants reduce friction by preventing actual contact of the metal surfaces, and substituting the lower friction of the lubricant itself.

Desirable Qualities of a Lubricant.—There are several important requirements a lubricant should possess:

1. Body;

2. Fluidity or viscosity;
3. Freedom from gumming;
4. Absence of acidity;
5. Stability under temperature changes;
6. Freedom from foreign matter.

The body of a lubricant indicates a certain consistency of substance, that prevents it being entirely squeezed out from the rubbing surfaces. The particles of the lubricant should adhere to the rubbing surfaces, thus securing effective separation. The body of a lubricant should be such as to prevent a too rapid running off, depending on the rubbing pressure.

FIG. 2,166.—***Cold test.*** This is indicated by the temperature at which oil ceases to flow. An extremely low cold test is obtained at the sacrifice of high flash test.

Fluidity of a lubricant refers to a certain lack of cohesion between its different particles, which reduces the fluid friction. Fluidity, so far as it does not oppose ***body,*** is a desirable quality. Excessive fluidity allows the lubricant to run off too quickly, thus causing waste.

A lubricant that gums loses its fluidity easily, collects dust and grit, and thus increases friction and wear generally.

A lubricant that holds free acid would attack the bearing surface, destroy its smoothness, increase friction, and lead to frequent and costly repairs.

Stability under temperature changes is important; lubricants should retain their good qualities, even when used under high temperatures as in a steam cylinder, or when used under low temperatures, as in ice machines, or on exposed bearings. They should not evaporate, not be decomposed by heat, nor congeal by cold and should retain their normal body and fluidity as much as possible.

Foreign matter will increase friction, and clog feed tubes, thus causing heating and possible seizing of the rubbing surfaces.

Cold, Flash, and Burning Points.—There are three critical temperatures of a lubricant which limit its application and which partly determine the conditions to which it is best suited.

Fig. 2,167.—*Flash and fire tests.* The ***flash test*** is indicated by that temperature at which the vapor given off by an oil ignites; the ***fire test,*** that temperature at which the oil itself burns. In making these tests the oil is heated in a small shallow vessel placed on a hot sand bath.

The cold point is the temperature at which any given grade of oil will either freeze or become cloudy.

The flash point is the temperature at which the oil gives off inflammable vapors.

The burning point is the temperature at which the oil takes fire.

Classes of Lubricants.—According to form or state, lubricants may be classified as:

1. Solid;
2. Liquid;

and with respect to the composition as:

1. Animal;
2. Vegetable;
3. Mineral.

Solid lubricants include, graphite, soapstone, and the various lubricating greases.

Graphite exists in two forms: *Crystalline* (or flake) and ***amorphous.*** It is also known as ***black lead*** and ***plumbago.*** Black lead usually refers to inferior grades of graphite, plumbago, to the Ceylon product, and graphite, to the American product. Graphite may be used alone or in combination with oil. The action of graphite is to fill the pores of the metal making the rough surfaces, shown in fig. 2,164, smooth, rather than to form an intervening film to prevent contact. Strictly speaking, graphite is not a lubricant, but in filling the pores of the metal it greatly reduces friction. One desirable quality of graphite when used in the cylinder is that its presence in a boiler does not produce any injurious effect.

Soapstone, also called *talc* or *steatite* is used as a lubricant in the form of a powder, or mixed with oil or fat. Mixed with soap, it is used on surfaces of wood working against either iron or wood.

The various lubricating greases are well adapted for heavy pressures under slow speed, but not for high speed, as their internal or fluid friction is considerable. The lubricating quality of grease may be improved by mixing with graphite. An advantage of grease is that it does not run, hence the machinery can more easily be kept in a clean condition.

Liquid lubricants are used extensively for both internal and external lubrications. They represent, in some forms, the highest quality of lubricants, having considerable body, with good fluidity and small internal friction. Their fluidity, however, may lead to large and even wasteful consumption, since by being easily spattered over all parts of the engine, they do not permit so great cleanliness as may be had with solid lubricants.

Animal oils such as, *sperm, whale, fish, lard* and *Neat's foot* *oils* are used to some extent. They are obtained by boiling or melting from the raw animal parts. As acid is sometimes used in the process of manufacture, animal oils are liable to have an acid reaction, and are then undesirable.

Sperm oil is an excellent lubricant; it does not become rancid, nor dry up; has good body and is fluid with little internal friction. It is used for rapid running parts, where a high grade is desirable, without much regard to price.

Whale oil is frequently used for external lubrication; it is a good lubricant at a moderate price.

Fish oil is also employed to advantage by some engineers for external lubrication.

FIG. 2,168.—***Gravity test.*** Gravity indicates the weight of the oil compared to water. Oil manufacturers use an arbitrary scale known as Baumé which gives it in values the reverse of the ordinary accepted specific gravity. The specific gravity of water being 1; any liquid lighter is less than 1, such as oil or gasoline, whereas on the Baumé scale, water being 10 all lighter liquids are greater than 10. High gravity is an indication of the quality of the crude oil from which the oil is refined.

Lard oil is used chiefly for mixing with other oils.

Neat's foot oil, on account of high price, is used in small quantities only for improvement of oils of poorer quality.

Vegetable oils are obtained by pressing the raw materials, and cleansing out the cloudy suspended fibres by treatment with acids. The color of the refined oils is from water, white to light yellow. Under heat they evaporate easily, and are, therefore, employed only for external lubrication. Vegetable oils are gradually decomposed by the oxidizing influence of the atmosphere, and dry up; they are also inclined to gum. Olive cotton seed, peanut, castor and rape oil are all used to some extent.

Olive oil is a good lubricant; it neither dries up, nor gums, but generally contains acid. On account of its high cost it is frequently adulterated with cheaper oils.

Cotton seed oil dries up less easily than others, and is consequently used sometimes as an admixture to olive oil. Certain grades frequently show an acid reaction, and are undesirable for lubricating purposes.

Linseed oil dries up easily, and is therefore undesirable for lubrication; it is often found as an adulterant in other oils on account of its cheapness.

Mineral oils are obtained by the distillation of petroleum; these oils are the most important lubricants, and since, with modern methods of manufacture, their price is relatively low. They retain their qualities well in the air, and if pure, do not gum or dry up.

The color for the different grades varies from light yellow to dark brown. The specific gravity ranges from 88 to 92, and the heavier oils are generally the thicker and less fluid. The body is generally higher with the thicker oils, comparing those of the same color. A compromise between body and fluidity must be made for the best and most economical result, so that good adhesion and slow running off is combined with moderate interval friction.

For cylinder oils the body should be the more pronounced quality, as fluidity is increased by elevation of temperature.

Mixtures of different oils are sometimes offered by manufacturers for cylinder lubrication; mineral oil as the main part is often mixed with sperm, whale or lard oil as a compromise between price, body and fluidity. Such mixtures should not be used inside a cylinder; only the best mineral oil should be used for this purpose.

Oil Tests.—There are two kinds of tests: *chemical*, and *mechanical*. The former are made in laboratories, but there are a number of simple tests which any engineer can make.

A test for clearness is made by taking a sample from a barrel that has been well rolled and shaken. The glass should be transparent, and the oil, if very cold, should be slightly warmed. The oil then, if of good quality, should be clear.

The amount of suspended matter is, with a light oil, directly seen, or for darker oils, determined by mixing and shaking with a relatively large quantity of gasoline.

The rancid test is made by mixing a small quantity of oil with warm water or alcohol, and testing with *blue litmus paper*, which will turn red if any free acid be present.

Rancid oil is indicated by the smell when a few drops are rubbed between the hands.

The Purity test.—This test is made by shaking a small quantity in a bottle, with a quick jerking motion, so as to produce air bubbles. If the oil be pure, the bubbles will soon burst and disappear, but if mixed with other oils, they will rise to the surface and collect.

Choice of a Lubricant.—There are several conditions that determine the choice of a lubricant for any given purpose, the principal things to be considered being:

1. Price;
2. Rubbing pressure;
3. Rubbing velocity;
4. Temperature.

It is generally supposed that the higher priced a lubricant is the better it is; this is correct theoretically, but not always so practically. The highest grade of lubricant is that which contains the highest percentage of lubricating matter, hence, it does not always pay to use the cheapest oil which can be obtained, for though the cheap oil costs less per gallon, the consumption to provide the same amount of lubrication is, in many cases, greater in proportion than the difference in price.

For heavy pressures a lubricant should have a good deal of body, while for lighter pressures there should be less body.

For high speed, a lubricant should, preferably, possess good fluidity, while, for slow speed, less fluidity is desirable to prevent waste.

Temperature has an important bearing on the choice of a lubricant; for instance, in the case of cylinder lubrication, it is evident that to prevent the lubricant being decomposed by the heat, the flash point should be higher than the maximum temperature of the steam in the cylinder.

For lubricating the various rubbing surfaces and turning joints, except within the cylinder, use is made of vegetable, animal, and the lubricating grades of mineral oil. In lubricating cylinders and internal surfaces nothing but the best grade of mineral oil should be used, the grade commonly employed being known as cylinder oil.

CHAPTER 41

LUBRICATION

How to Oil an Engine.—The subject of lubrication should receive the special attention and study of every engineer. It is quite important that the engine be properly oiled to avoid excessive friction, wear, and trouble. A small amount of a well selected oil properly applied will go further in reducing friction than a much greater amount of an unsuitable lubricant improperly applied. Oiling an engine involves:

1. Internal lubrication; and,
2. External lubrication.

The former includes oiling the cylinder and valves, and the latter, the external bearings.

Internal Lubrication.—There are several kinds of lubricator for introducing oil into the cylinder; these may be classified with respect to their principles of operation as:

1. Gravity;
2. Hydrokinetic;
    - *a.* **Up flow;**
    - *b.* **Down flow.**
3. Force feed.

Gravity Lubricators.—Those working on this principle are called "plain lubricators," there are two tubes, the invisible feed, and the sight feed. The action of gravity lubricators depends on: 1, the displacement of the oil from the reservoir by condensation, and 2, its movement downward by gravity.

Fig. 2,169 is a sectional view of a plain lubricator with invisible feed. ***In operation,*** steam passes through the central tube to the upper part of the oil reservoir, where it condenses. The water thus formed being heavier than the oil, sinks to the bottom, displacing a corresponding amount of the oil which overflows through the open top of the tube, and descends by gravity to the engine.

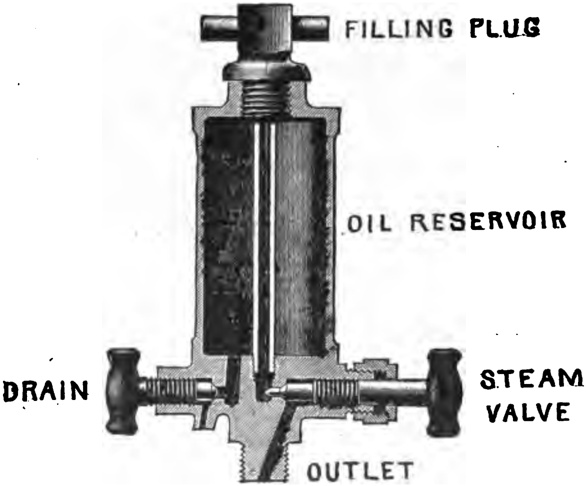


FIG. 2,169.—Nathan plain cylinder lubricator. ***In operation,*** steam passes up the central tube and condenses in the oil reservoir, displacing the oil which flows over the top of the tube, and down to the cylinder.

hi **Hydrokinetic Lubricators.***—The operation of lubricators

Fo is class depends on two well known principles of physics:

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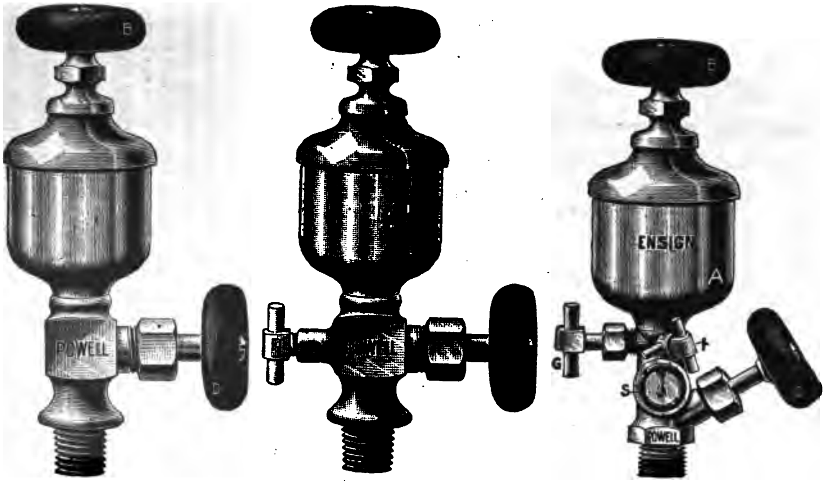
E.—The word *hydrokinetic* is defined as: "Relating, or pertaining to the motions of is applied to this class of lubricator, whose operation is due, primarily, to the down- g of an elevated body of water which displaces the oil from the reservoir.



FIGS. 2,170 to 2,172.—Powell plain cylinder lubricators; fig. 2,170, plain lubricator without drain. The valve in the shank makes it possible to fill the cup while the engine is running and also acts as a feed regulator; fig. 2,171, plain lubricator with condensing tube, oil regulating valve and draw back; fig. 2,172, sight feed lubricator with single connections.

1. If a body (the oil) be acted upon by two unequal pressures it will move in the direction of the greater force.

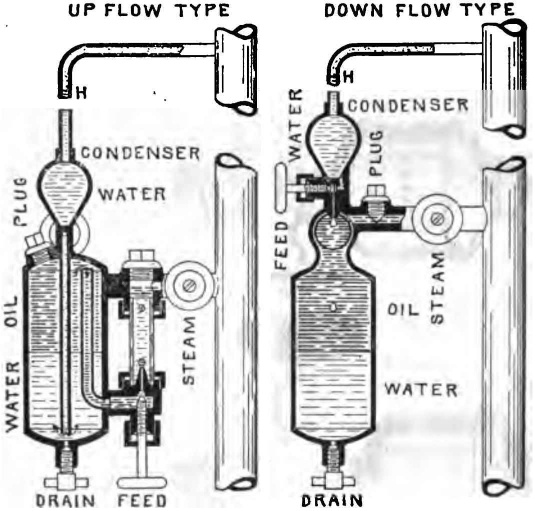
2. The specific gravity, or the weight of a certain quantity of oil is less than the same quantity of water, hence the oil will rise to the top.

With these facts in mind, the operation of any hydrostatic lubricator may be easily understood. There are two forms of hydrokinetic lubricator as shown in figs. 2,173 and 2,174, and known as the up flow and down flow types. In the former, the oil is visible, rising drop by drop in the sight glass, while in the latter drops of water are seen descending to the bottom of the oil reservoir.

The principal parts of an up flow lubricator as shown in fig. 2,173, are: 1, condenser; 2, oil reservoir; 3, tube connecting the condenser to the lower part of the reservoir; 4, tube connecting upper part of the reservoir to, 5, the sight feed.

Steam from the main steam pipe passes into the connecting pipe above the lubricator, and condenser, filling the condenser and part of the pipe above it with water to some height as H. When the steam valve is opened, the sight feed glass is also filled with condensation.

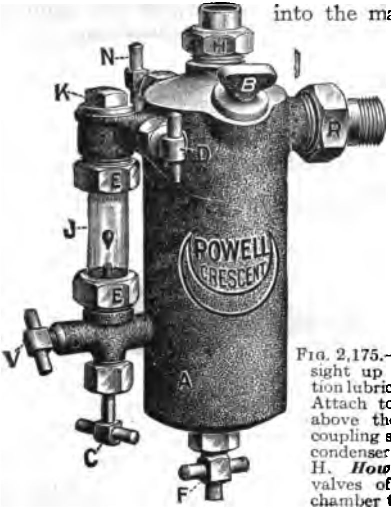
In operation, when the condenser and steam valves are open, water from the condenser will pass down the central tube to the lower part of the reservoir, and being heavier than the oil, will stay at the bottom, the oil floating above. On account of the excess pressure in the condenser tube due to the head H, the water will continue to flow until the oil fills the upper part of the reservoir. When the feed valve is opened, the excess pressure due to the head of water will force the oil, drop by drop, through the nozzle in the sight glass. As soon as a drop of oil leaves the nozzle, it is no longer acted upon by this excess pressure, but rises because it is lighter than the surrounding water in the sight glass.



FIGS. 2,173 and 2,174.—***Up flow,*** and ***down flow*** hydrokinetic lubricators. In the up flow type, fig. 2,173, the oil ascends, drop by drop, through the sight glass, while in the down flow type, fig. 2,174, descending drops of water are visible through the glass sight discs. The operation of hydrokinetic lubricators depends on: 1, an excess pressure produced by a head of water, and 2, a difference in density between the water and the lubricant.

In a down flow lubricator there are no internal pipes connecting with the condenser and sight feed. The sight glass consists of two glass discs inserted in the upper part of the reservoir as shown in fig. 2,174.

The operation is quite simple: When the condenser needle valve is opened, the water from the condenser will flow through the passage, and, as can be seen through the sight discs, leave the nozzle drop by drop. It being heavier than the oil descends to the bottom displacing an equal amount of oil which is discharged into the main steam pipe.



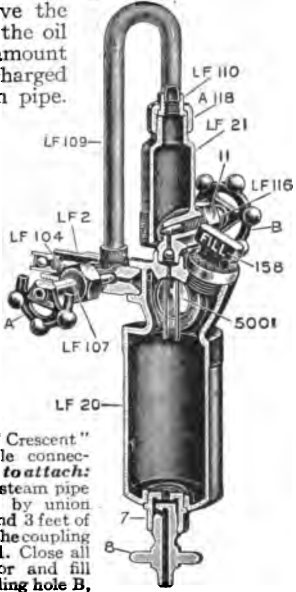


FIG. 2,175.—Powell "Crescent" sight up flow double connection lubricator. ***How to attach:*** Attach to vertical steam pipe above the throttle by union coupling shank R, and 3 feet of condenser pipe with the coupling H. ***How to use:*** 1. Close all valves of Lubricator and fill chamber through filling hole B, with clean oil and replace cap; 2. Open wide steam valve D, and water valve N, and wait until sight glass J, fills full of water before starting the cup. 3. Regulate oil drops by valve C, at bottom of sight chamber to desired rate of feed. To prevent pulsation, close valve N, more or less, as required. ***To replace sight glass,*** if broken, remove cap K, and insert glass, leaving a little end play to allow for expansion. ***To refill,*** draw off water through drain valve F, and proceed as before. Clean sight glass by blowing steam through vent cock V. ***To prevent freezing,*** be sure to drain the lubricator. Draw off any unused oil and condensed water by opening valves N and F, drain plug V, and filling cap B.

FIG. 2,176.—Detroit "400" sight feed lubricator (down flow type). ***The parts are:*** 5,001, glass; LF2, support arm; LF21, condenser; LF104, oil tube; LF4, feed valve pkg. rings, not shown; LF107, oil feed valve stem complete; LF116, water feed valve stem complete; 11, packing nuts; 158, filler plug; LF110, tail pipe (single connection only); A118, tail nut; 8, drain valve body; 8, drain valve stem; oil tube ball check; 23, plug for tube hole, in double connection, not shown; LF105, tail pipe for double connection, not shown; LF109, equalizing tube (single connection only); water feed ball check; LF20, body for ¼ pint; LF22, body for ⅛ pint; LF23, body for ½ pint; LF24, body for 1 pint; LF25, body for 1 quart.

Practical Points.—Engineers experience more or less trouble in the daily operation of lubricators from one cause or another. To avoid this, the foregoing principles should be clearly understood.

Before starting a lubricator, time should be allowed for the condenser and sight feed glass to fill.

A common fault is fouling of the sight glass. This is usually due to the condition of the nozzle; it frequently becomes covered with dirt and sediment from the oil, which makes the surface rough, causing the drops to adhere too long to the nozzle. This condition causes the drop to become so large that it strikes the side of the glass in rising, thus gradually covering the glass with particles of oil which become detached from the drop at each contact. This may be overcome by removing the glass and cleaning the nozzle both inside and out, rubbing it smooth with crocus cloth.

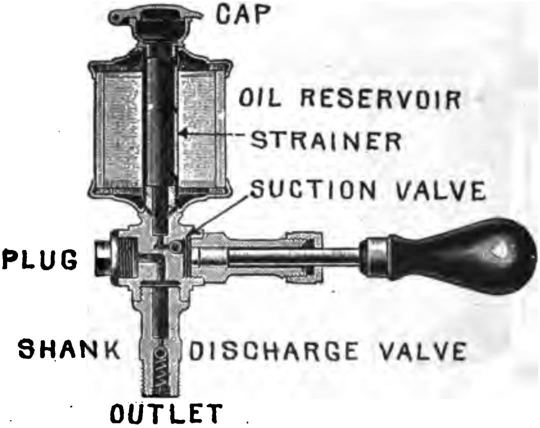


FIG. 2,177.—Nathan hand oil pump; for occasional use in cylinder lubrication. It may be attached either vertically or horizontally by interchanging the shank and plug. A valve should be placed on the outlet, and should be kept closed when the pump is not in use.

Sometimes the orifice in the nozzle is large for the kind of oil used. This causes large drops to form, which tend to foul the glass.

A lubricator should be blown out occasionally so as to remove any dirt or sediment that may have accumulated in the small tubes and passages.

When the engine is shut down, as during the noon hour, and the feed valve is closed, the condenser valve in the up flow type, should be left open. If both valves are shut there will be no outlet, hence, if the temperature of the oil should rise, *it will expand and exert such a pressure on the reservoir as to cause it to bulge or burst.*

Force Feed; Oil Pumps.—Hydrokinetic lubricators are affected by changes in temperature, causing them to feed too slow in cold weather and too fast in warm weather. In an effort to overcome this defect, what is known as force feed, or oil pump lubricators have been designed and put on the market with more or less claims as to their ability to provide positive and uniform lubrication.

There are two kinds of force feed lubricator: 1, the hand pump which is used as an auxiliary to the main lubricator, and 2, the power pump operated by the engine, and employed as the regular feed.

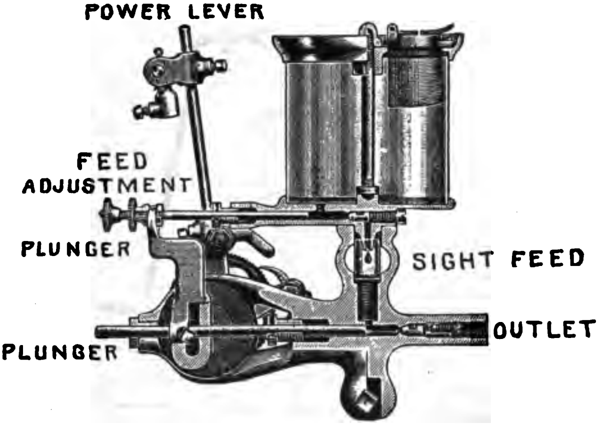


Fig. 2,178.—Manzel power sight feed oil pump. The oil is forced out of the reservoir and through the sight glass by the upper plunger, and is then forced on through the check valves and into the cylinder by the lower plunger. The amount of oil supplied with each stroke is regulated by the adjustable upper plunger.

Fig. 2,177 shows the ordinary hand oil pump. The oil reservoir has a removable strainer inserted in the central tube, and the filling hole is covered by a cap to keep out dust and impurities. The pump is of the single acting plunger type with ball valves as shown. By reversing the positions of the plug and shank, the lubricator may be adapted to horizontal connection.

There are many designs of force feed pump, one being shown in fig. 2,178.

In operation, oil is drawn from the reservoir and forced through the sight glass by the upper plunger; it is then forced on through the check valves and outlet by the lower plunger. The amount of oil supplied with each stroke is regulated by the adjustable upper plunger. The lower plunger is made slightly larger than the upper, to avoid any possibility of oil remaining in the sight glass. Motion is imparted to the plungers by means of a ratchet wheel and cam, which in turn are moved by a lever connected to some reciprocating part of the engine. There is a hand attachment on the ratchet wheel to permit hand operation before starting the engine, or when more oil is needed momentarily while the engine is running.

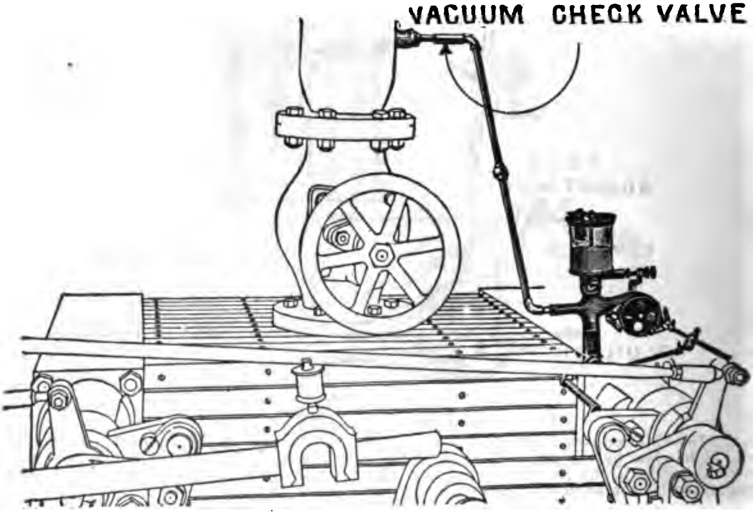


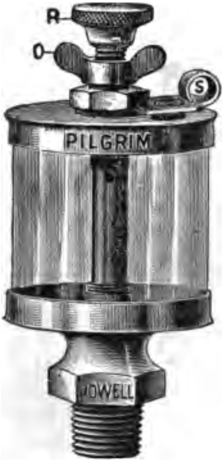
Fig. 2,179.—Method of attaching a power oil pump to cylinder. A vacuum check valve is placed at the end of delivery pipe to prevent the oil being siphoned out of the reservoir in case a vacuum form in the boiler or cooling. Motion for operating the plungers is obtained from some convenient part of the valve gear.

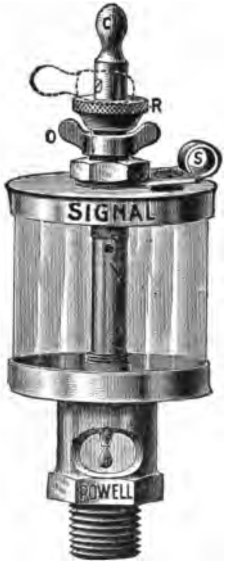
Fig. 2,179 shows method of attaching pump to a Corliss cylinder. It should be noted that a spring check valve is placed on the oil discharge pipe to prevent the oil being drawn out by the vacuum which forms in the boiler on cooling.

External Lubrication Systems.—The successful lubrication of the bearings of an engine depend in a measure upon the character of the appliances used to convey the lubricant to the wearing surfaces. There are several systems of external lubrication, the choice of which is governed by the type of engine, and conditions of service.

They may be classified as:

1. Gravity;
2. Inertia;
3. Centrifugal;





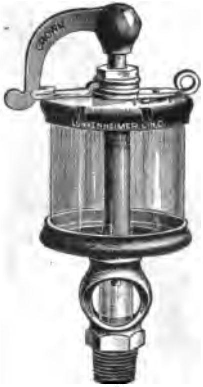


Fig. 2,180.—Powell *"Pilgrim"* plain glass oil cup. The feed is regulated by the milled screw feed stem R, and is secured by the winged jamb nut O.

Fig. 2,181.—Powell *"Signal"* snap lever sight feed oil cup. The feed is turned on or off by the snap lever C, being *on* in the vertical position and *off* in the horizontal position. The rate of feed is regulated by the milled screw feed stem R, and secured by the wing nut O.

Fig. 2,182.—Lunkenheimer *"Crown"* index sight feed oil cup. The index device is for regulating the feed, and the indicator arm turning on the lid, marks the notch giving the desired feed.

4. Capillary { wick feed; chain, and collar feed;
5. Pressure;
6. Compression;
7. Splash.

Gravity System.—In this method of oiling, the lubricator is placed at a sufficiently high elevation to permit the oil to gravitate or flow to the bearing. Many of the sight feed cups work on this principle. These cups are made in single, or multiple units as shown, figs. 2,180 to 2,183.

Fig. 2,180 shows a plain oil cup, an inexpensive though satisfactory device for some feeds. The cup shown in fig. 2,181 is of the snap lever type with sight feed. This is a very desirable type as the feed is visible and can be turned on or off without changing the feed adjustment. By tilting the snap lever half way between the *on* and *off* positions, the bearing may be flushed with oil.

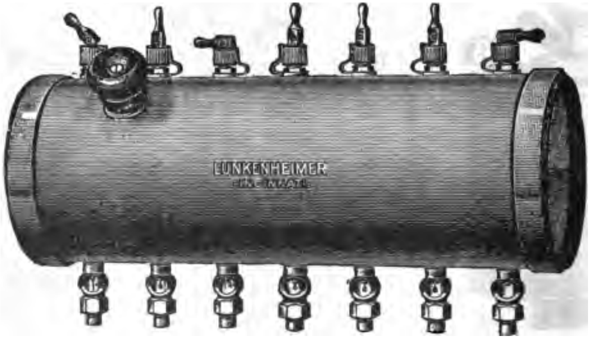


FIG. 2,183.—Lunkenheimer multiple sight feed oiler. This consists of a number of sight feeds of the "Sentinel" type placed in a common reservoir. A union is placed at the end of each feed for easy connection with the oil pipes. Each end of the cylindrical reservoir is closed with a glass disc making the supply of oil visible.

Fig. 2,182 shows an "index" sight feed cup. The feed is regulated by an index device, and an indicator arm which turns on the lid to mark the notch giving the desired feed. When the feed is shut off, the lever is placed in a vertical position to indicate that the cup is not working.

The multiple oiler shown in fig. 2,183 consists simply of a number of sight feeds having a common reservoir. The ends of the reservoir are of glass to indicate the amount of oil inside. Each sight feed terminates in a union to which tubing is attached; these lead to the various bearings.

Figs. 2,185 to 2,192 illustrate a few gravity oiling devices of which there is a large variety.

Capillary Systems.—These include wick feed lubricators, and the chain, and collar devices used in self oiling bearings.

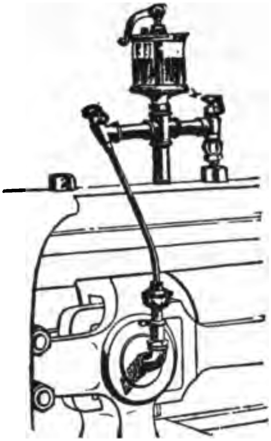
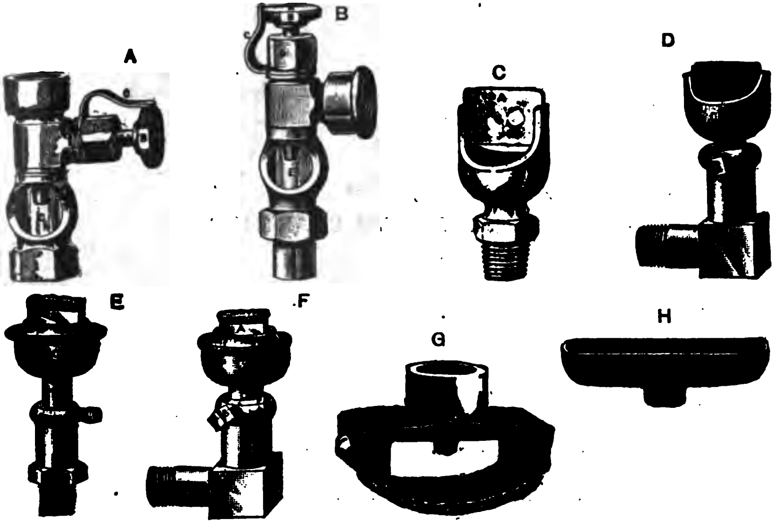


FIG. 2,184.—Oil cup with two branches for lubricating wrist pin and guide. A wiper cup is placed on the cross head, and an angle sight feed on the guide.



FIGS. 2,185 to 2,192.—Lunkenheimer oiling devices. A, is an oil regulating valve with sight feed; B, angle oil regulating valve with sight feed; C, wiper cup; D, angle wiper cup; E, wiper; F, angle wiper; G, wiper; H, drip cup.

Wick feed lubricators are provided with one or more small tubes, tapped, oil tight, into the bottom of the reservoir, and reaching to the top as shown in fig. 2,193. To lift the oil automatically over the top of the tube a wick is inserted with one end dipping into the oil in the reservoir. The end of the tube and wick must project below the bottom of the reservoir. A wire is attached to the wick to hold it in place. The wick, due to capillary attraction, becomes saturated with oil, which is siphoned from the reservoir drop by drop. This system is used extensively on marine engines, and although very reliable, the rate of feed cannot be regulated so easily as the gravity feed.

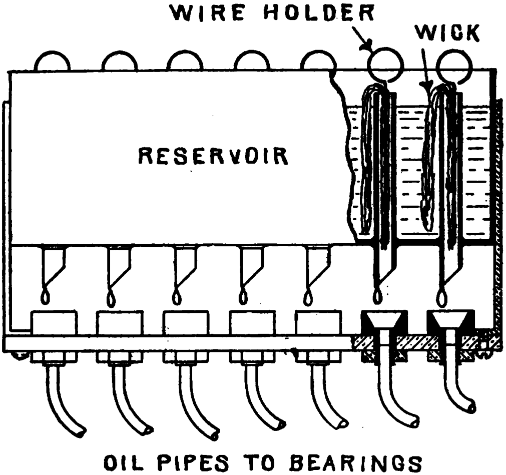


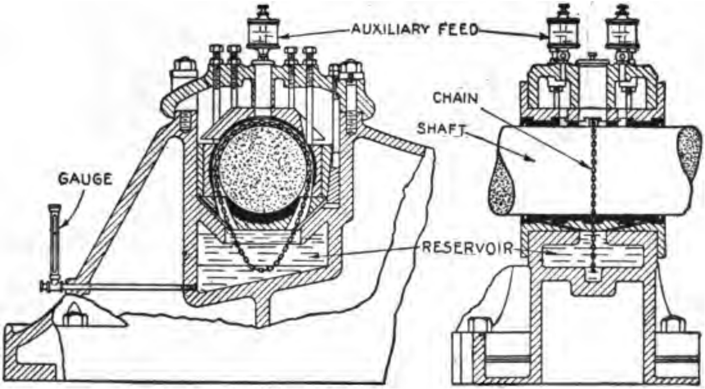
FIG. 2,193.—Multiple wick feed oil cup. A number of tubes are tapped oil tight through the bottom of the reservoir; these extend above the oil level, and have a wick inserted in each with one end dipping in the oil. It soon becomes saturated by capillary action and is then siphoned drop by drop. A wire is attached to each wick so that they may be easily inserted or removed from the tube.

An endless chain or collar is used on what is known as a "self oiling" bearing as shown in figs. 2,194 and 2,195.

The length of the chain is such that it dips into an oil reservoir directly under the bearing and in rotating with the shaft, the ascending side carries with it, by attraction between the liquid and metal, a small quantity of oil which lubricates the shaft.

Inertia System.—Lubricators which operate on the force due to inertia are adapted to oiling a reciprocating part moving in an up and down direction, as the wrist pin of a vertical engine. Two types of inertia cups are shown in figs. 2,196 and 2197.

In fig. 2,196, the central tube connecting with the outlet contains a plunger feed valve, which by its inertia opens or closes at each end of the engine stroke, thus acting as a piston in forcing the oil to the bearing at each revolution.



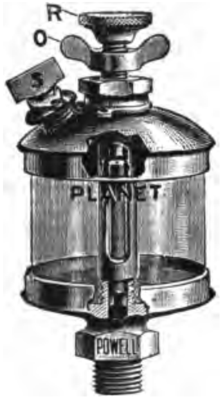
Figs. 2,194 and 2,195.—Sectional views of chain oiling bearing of the Russel engine. An endless chain encircles the shaft and dips into the oil reservoir beneath. As the chain moves with the shaft, oil clings to the ascending side, and is thus carried to the top of the shaft for lubrication; any excess drains back to the reservoir through the passages in the lower portion of the bearing.

The amount of opening or stroke of the plunger valve is adjusted by the milled nut R, and set by the jamb nut O.

In the second type fig. 2,197, the valve is made stationary and the feed obtained by the inertia of the oil itself.

In this cup the oil is thrown against the top at the beginning of the down stroke and enters the central tube. The rate of feed is adjusted by the valve R.

Centrifugal System.—For oiling crank pins lubricators are sometimes used which depend on centrifugal force for their operation. There are several types as shown in figs. 2,196 to 2,201.



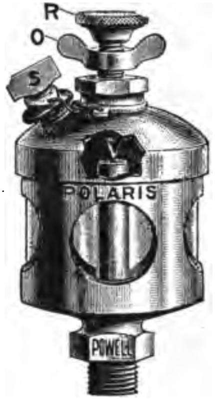
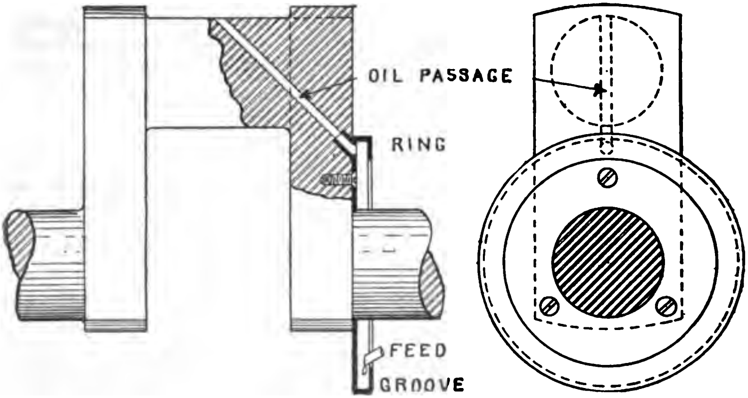


Fig. 2,196.—Powell "Planet" plunger feed crank pin oil cup. The plunger feed valve acts as a piston, focing the oil to the bearing at each revolution.

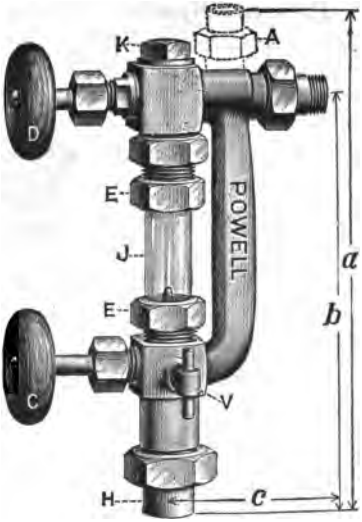
Fig. 2,197.—Powell "Polaris" crank pin oil cup with needle point feed. This type of feed permits nicety of adjustment, at the same time insuring a positive feed for crank or wrist pins.

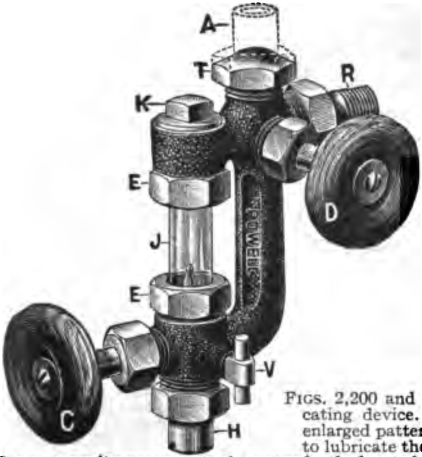


Figs. 2,198 and 2,199.—Ring centrifugal crank pin oiler. This type is adapted to center crank engines. An oil passage leads from the grooved ring to the crank pin. In operation, the oil, which drops into the groove, is carried off by centrifugal force through the oil passage to the crank pin. ***In construction,*** the oil passage should be of liberal size to prevent clogging.

In fig. 2,198 a grooved ring is attached to the crank to the oil from the sight feed. There is a connecting passage or duct through which the oil may flow from the groove to the crank pin bearing. *In operation,* centrifugal force tends to throw the oil from the center of rotation, hence it presses against the bottom of the groove and is forced through the duct to the bearing, thus lubricating the pin. This type of oiler is used mostly on center crank engines.

For a side crank engine, the pendulum bob type of oiler is extensively used.





FIGS. 2,200 and **2,201.—Powell individual sight feed lubricating device. Fig. 2,200, standard pattern; fig. 2,201, enlarged pattern. It is intended for use when it is desired to lubricate the cylinders of two or more engines from one large capacity pressure tank conveniently located. The supply being taken from the top of the tank, and by means of ¼ inch pipe conducted to the device and connected at coupling H. The attaching shank R (⅜ pipe) is screwed into the steam pipe. C, is drop regulating valve, V, blow off valve for cleaning sight chamber. K, is removable plug through which the sight glass is replaced. For compound engines it is best to use an equalizing pipe connected to steam pipe above the throttle. When so used, bonnet T, is replaced with a union coupling. Using this device saves the time ordinarily required in filling the sight feed lubricators on each engine.**

The tube or arm through which the oil flows by centrifugal force to the crank pin is bolted to the end of the crank pin, the bearing at the other end which carries the cup is concentric with the center of the shaft, thus the oil cup, which is here journaled, and weighted with a pendulum bob, remains stationary, and in an upright position. In operation the oil drops from the end of the inner tube, and is carried by centrifugal force to the crank pin.

The cup is circular in section and has glass sides to indicate the amount of oil. The feeding arrangement consists of a tube screwed into the base and communicating with the outlet. At the top is a regulating valve. The cup is placed in an upright position at the end of the connecting rod. In operation the centrifugal force due to the rotary movement of the cup, throws the oil outward against the circular walls and the feed inlet. It is obvious, no matter how little oil there may be in the cup, it will be carried to the feed inlet.

Pressure System.—In this method of lubrication the necessary pressure for forcing the lubricant to the bearings may be obtained 1, by placing the reservoir at a suitable elevation; 2, by employing air pressure in a closed reservoir, or, 3, by means of a force pump. Figs. 2,203 and 2,204 show one method of pressure feed.

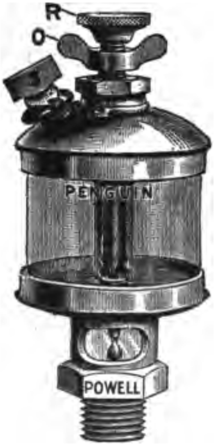
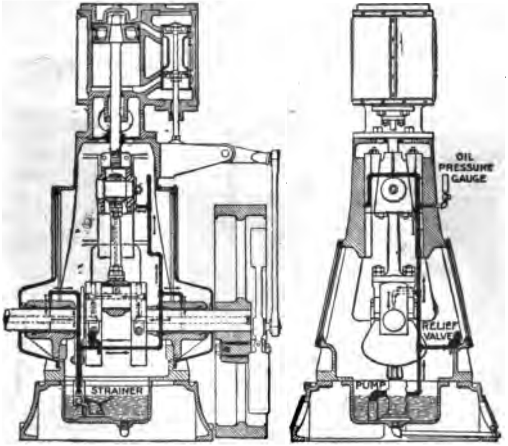


FIG. 2,202.—Powell "Penguin" slide sight feed oil cup designed for use on slides, cross heads, eccentrics, and all moving bearings. The feed is easily set by regulating feed stem R, and secured by jamb nut O. The filling plug is attached to cup by a small chain to prevent its getting lost.

An inclined receptacle forming a part of the bed plate contains the oil and has attached a small plunger pump operated by an eccentric on the shaft. The oil is distributed through feeds to the various bearings; the frame is enclosed, hence there is no loss of oil in being thrown off from the bearings. It is thus used over and over again, however it should be noted that with systems of this kind the oil should be frequently filtered and renewed.

Compression System; Grease Cups.—Grease is frequently used instead of oil on some bearings, especially those which run slowly and with considerable pressure. The numerous kinds of cup used for the application of grease may be classed as:



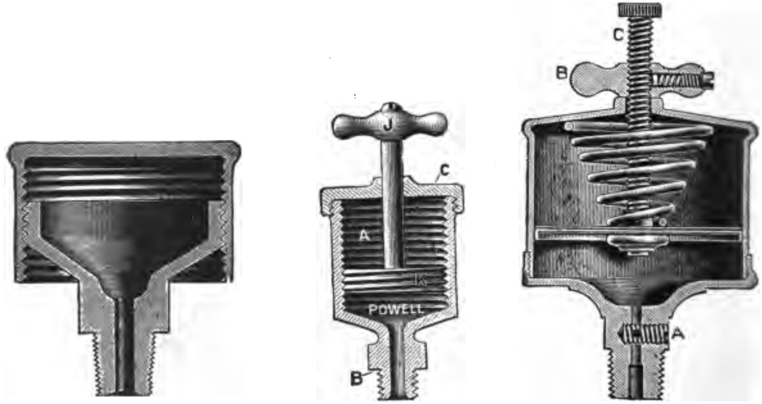
Figs. 2,203 and 2,204.—Sturtevant vertical engine. Side, and end views showing pressure system of lubrication. The frame is enclosed, and at the bottom is an oil reservoir and pump which is operated by the engine. This pump forces the oil through a system of pipes to all the bearings from which it drains back to the reservoir.

1. Hand operated;
2. Automatic.

All grease cups operate by compression.

Fig. 2,205 shows a simple form of cup. The lubricant is fed to the bearing by screwing down the cap over the stationary bottom part, thus compressing the grease and forcing it through the outlet.

A screw piston or "marine type" cup is shown in fig. 2,206; this is a desirable cup in places where it is necessary to force the grease some distance to the parts to be lubricated. The body of the cup is threaded on the inside, and the piston to correspond. By turning the handle attached to the piston the grease is compressed and forced out of the cup. Fig. 2,207 shows an automatic cup. It is provided with a leather piston which is easily raised by a thumb nut, whenever re-filling is necessary. The thumb nut which controls the spring and piston, has a locking arrangement, which prevents it jarring from its position on the main stem. The feed is regulated by means of a screw plug in the shank of the cup.



Figs. 2,205 to 2,207.—Hand and automatic compression grease cups. Fig. 2,205 shows a simple cup, and fig. 2,206, a "marine type" with screw piston. The automatic cup shown in fig. 2,207 is provided with a leather piston which may be raised by means of the thumb nut B, for refilling. The thumb nut, which controls the spring and piston, has a locking arrangement which retains it in position, on the stem C. The feed is regulated by means of the screw plug valve A.

To fill the cup, thumb nut B, is turned to the right which draws the piston to the top; the cup is then unscrewed from the base and filled with grease. After replacing, pressure is put upon the grease by screwing thumb nut B, to the top of the stem C, thus allowing the piston to be pressed downward by the spring. The rate of feed is regulated by means of plug A, which is drilled to register with the feed passage, according to the position of the plug.

If it be desired to stop lubrication while preserving the rate of feed, thumb nut B, is turned down to the top of the cup, thus preventing the further advance of the piston.

Splash System.—With this method of lubrication an enclosed frame is necessary. A quantity of oil is placed in the frame and maintained at such a level that the end of the connecting rod comes in contact with it at the lower part of the revolution, thus *splashing* it upon the working parts. This system is frequently used with high speed horizontal engines as shown in fig. 2,208. The amount of oil that is splashed over the bearings will depend

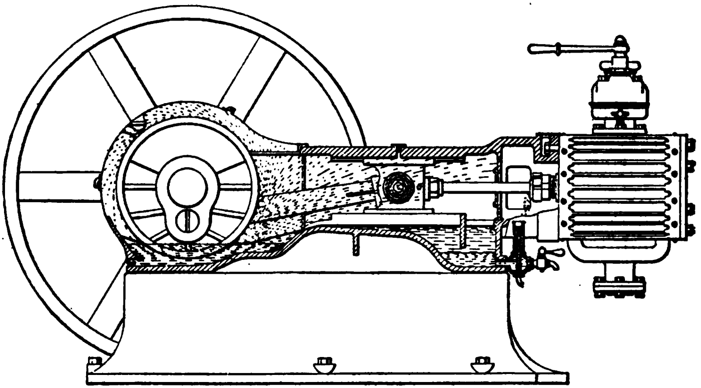


FIG. 2,208.—The McEwen (Ridgway) engine; sectional view showing splash system of lubrication. Oil is placed in the enclosed frame in sufficient quantity to submerge the connecting rod at its lowest point of travel, causing it to splash the oil over the various bearings. It should be noted that the piston rod stuffing box is outside the enclosed portion of the frame thus keeping the oil separate from any water of condensation, which may leak past the stuffing box—an important point in this system.

on the rate of speed and the depth of the oil in the frame. As with the pressure system the oil should be frequently filtered and renewed.

Practical Points on Lubrication.—The engineer should give special attention and study to the subject of lubrication, not only to avoid waste, but as a safeguard against the possibility of a shut down. An engine running up to speed no matter whether it be slow, medium, or high speed, should have the constant attention from the man in charge.

The temperature of the engine room has much to do with the rate of feed of oil cups and cylinder lubricators; the higher the temperature the faster will the oil flow.

Where two oil cups are used on a long bearing they should be placed not further on each side of the center than 1/6 the length of the bearing.

Oil grooves, after being cut in a bearing should have the edges rounded with a scraper to let the oil follow the shaft; sharp edges have a tendency to scrape off the oil.

Oil should be applied to the shaft at some point outside the area upon which the pressure comes. This can be done by the proper arrangement of the oil grooves.

The amount of oil required for a bearing should be determined with care by repeated trials; where there are no drip pans, it is important that the bearing receive no more oil than necessary, to avoid waste and to keep the engine room clean.

CHAPTER 42

MANAGEMENT

The term management, as here used, embraces, in addition to the attention given to the engine, the adjustment and control of the auxiliary apparatus connected with the engine, such as steam separator, oiling devices, condenser, condenser pump, etc. Management includes, therefore

1. Operation;
2. Care;
3. Repair;
4. "Laying-up."

The successful engineer must not only understand the necessary conditions of working and control, but he must know how to meet the numerous disorders and mishaps that may be encountered, as those arising

1. From faulty construction;
2. From careless or ignorant handling, such as,
    - *a.* Insufficient lubrication;
    - *b.* Faulty adjustments;
    - *c.* Racing;
    - *d.* Overheating;
    - *e.* Breakage or wear of parts.

1. OPERATION

Before Starting.—All oil cups and lubricators should be filled with the proper lubricant. If the steam pipe have a separator it should be drained. The cylinder relief cocks should next be opened and if there be a by pass, the cylinder may be warmed by slightly opening the valve; in the case of a reversing engine, the operation of warming the cylinder may be facilitated by moving the reverse lever back and forth several times while the by pass valve is cracked. In the case of condensing engines with independent pumps the air pump should be started before warming the cylinder.

Starting.—In starting an engine of any type, the main thing to **guard against** is *excessive condensation* in the cylinder, because of the danger of injury to the cylinder head and moving parts as the piston approaches the end of the stroke.

Water being an unyielding substance, when it fills the clearance space and is acted upon by a fast approaching piston, causes a shock to the cylinder head and moving parts as well, and in extreme cases even though the relief cocks be open there is danger of injuring the engine. Accordingly, in starting any engine, *steam must be admitted* **very gradually** *and the engine* **slowly** *brought up to speed.*

In the progressive opening of the throttle a knock at the end of each stroke indicates water hammer due to excessive condensation, hence, if this occur, gradually close throttle, or do not open any wider until the knocking ceases.

After the engine has been brought to speed, the relief cocks may be gradually closed.

In closing them, if a knock be heard, it indicates that the cocks have been closed too soon and must be again opened until all knocks or water hammer ceases. There are several cases of starting a condensing engine which must be considered, as considerable damage may be done in some instances unless the various operations are properly performed.

Starting with Direct Connected Air Pumps.—Frequently the condenser air pump is direct connected to the engine, hence there will be no vacuum until after the engine has started. One characteristic of this arrangement is the liability of flooding the condenser, hence in warming up, after starting the independent circulating pump, the engine should be turned over or, in the case of reversing engines, worked back and forth slowly as soon as possible so as to set the air pump in motion and clear the condenser of water.

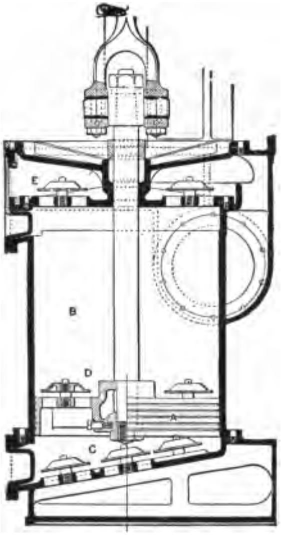


FIG. 2,209.—Direct connected vertical air pump. ***The lettered parts*** are: A, bucket or piston; B, barrel; C, foot valves; D, bucket valves; E, head valves.

In the case of compound and triple expansion engines ample time should be given to warming the cylinders, and working out the water. This is facilitated by the steam by pass valves to the receivers. It is frequently advisable to run multi-cylinder engines very slowly for several minutes before full speed operation.

With independent pumps they are started before warming up or starting the engine, each being adjusted to its proper speed.

Sometimes both circulating and air pumps are separate units requiring individual adjustment, but where the two pumps are direct connected to one power cylinder, the speed must be governed by the degree of vacuum desired.

With surface condensers where the condensate is returned to the boiler, the feed pump, if independent, must be started when starting the engine, otherwise the water pumped from the condenser by the air pump will flood the hot well and overflow, thus losing some of the condensate—an important item in marine plants where evaporators are not provided.

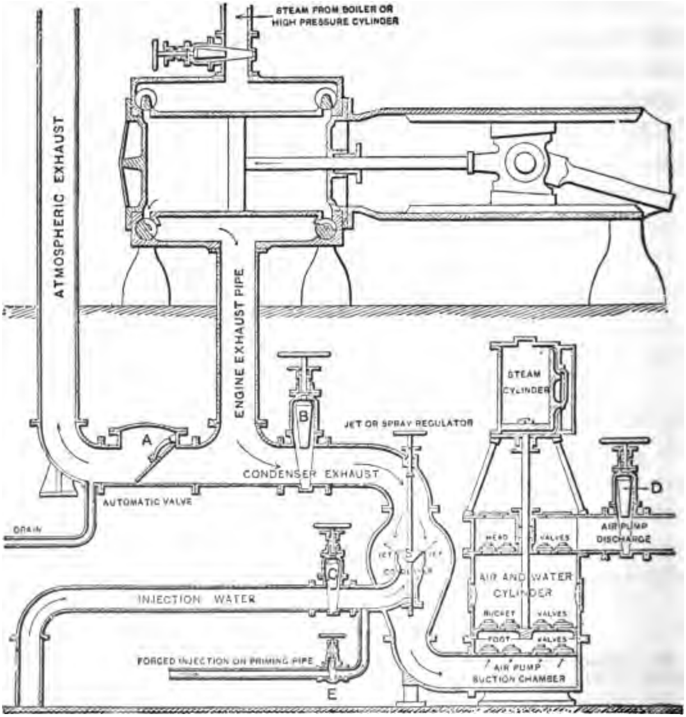


FIG. 2,210.—Diagram of Corliss cylinder and connections with Blake jet condenser, independent pump.

When the water tanks are below the level of the hot well an overflow pipe connecting the hot well should be installed to avoid loss of condensate by flooding.

Starting with Jet Condenser.—In the jet condenser, the air pump removes not only the air but the circulating water as well, hence it must be evident that if the pump do not remove the water from the condenser as fast it it comes in, the apparatus will quickly flood, unless the vacuum be broken by the proper working of an automatic device provided for that purpose, and back up into the engine cylinder causing serious damage. Hence in starting jet condensing, great precautions should be taken that the water does not reach the cylinder.

***To start:** open slightly the injection valve and start up the air pump to its normal speed.* This produces a vacuum in the pipes and condenser, drains them of water, and causes the injection of water into the condenser. *When the vacuum is established, as indicated by the gauge, slightly open throttle, and warm up engine. Then bring engine up to speed and regulate the amount of injection water by the injection valve.* In regulating the amount of injection water be guarded by the vacuum gauge: *if an increase of injection water does not cause an increase in vacuum, the speed of the air pump must be increased. The injection valve should never be opened further than will cause the vacuum to increase.*

Starting with Siphon Condenser.—Since in this arrangement the hot well is located 34 feet below the condenser no air pump is required to remove the condensed steam and cooling water. Hence, there is no way in which water can get into the engine cylinder unless it be allowed to accumulate in the pocket formed by the exhaust pipe, and not even then unless atmospheric pressure be admitted to the exhaust pipes through the uncovering of the water supply or discharge pipes; a proper drain will protect the exhaust pipe. Accordingly, starting with a siphon condenser is accompanied by no such danger, as with a jet condenser.

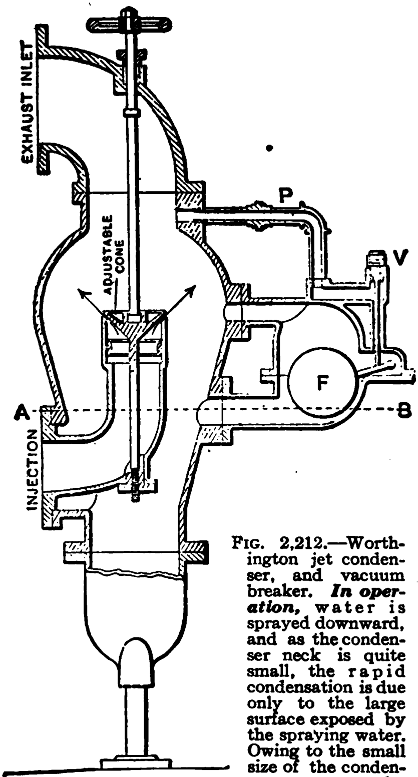


FIG. 2,211.—Blake jet condenser with vacuum breaker. ***In operation,*** water rises in the condenser to the level A B, and lifts the float F, which in turn lifts the air valve V, from its seat, admitting air to the exhaust pipe and engine cylinder through the pipe P, thus breaking the vacuum and stopping the flow of cooling water into the condenser. The engine exhaust will then raise the pressure sufficiently to lift the relief valve and escape into the atmosphere.

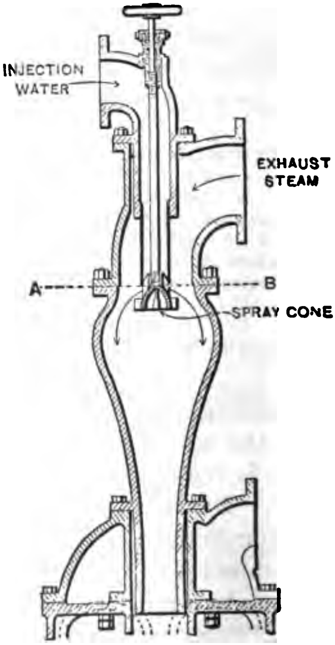


FIG. 2,212.—Worthington jet condenser, and vacuum breaker. ***In operation,*** water is sprayed downward, and as the condenser neck is quite small, the rapid condensation is due only to the large surface exposed by the spraying water. Owing to the small size of the condenser any accumulation of water rapidly diminishes the condensing surface until the spray itself is submerged, leaving only the small annular ring of water at A B to act on the large volume of steam from the engine. The surface of this ring is far too small to condense the steam and the pressure immediately accumulates and either the relief valve opens, allowing the engine to run non-condensing, or the exhaust steam blows out through the injection pipe and pump valves.

NOTE.—*In jet condensing,* the engine itself may draw water into the cylinder, thus in starting, say, a compound with very light load, the terminal pressure in the low pressure cylinder may be considerably below atmospheric pressure, then the vacuum thus produced will draw the water into the cylinder unless: 1, the water be carried away by the pump, or 2, the vacuum be broken by some form of vacuum breaker which breaks the vacuum by opening an air valve.

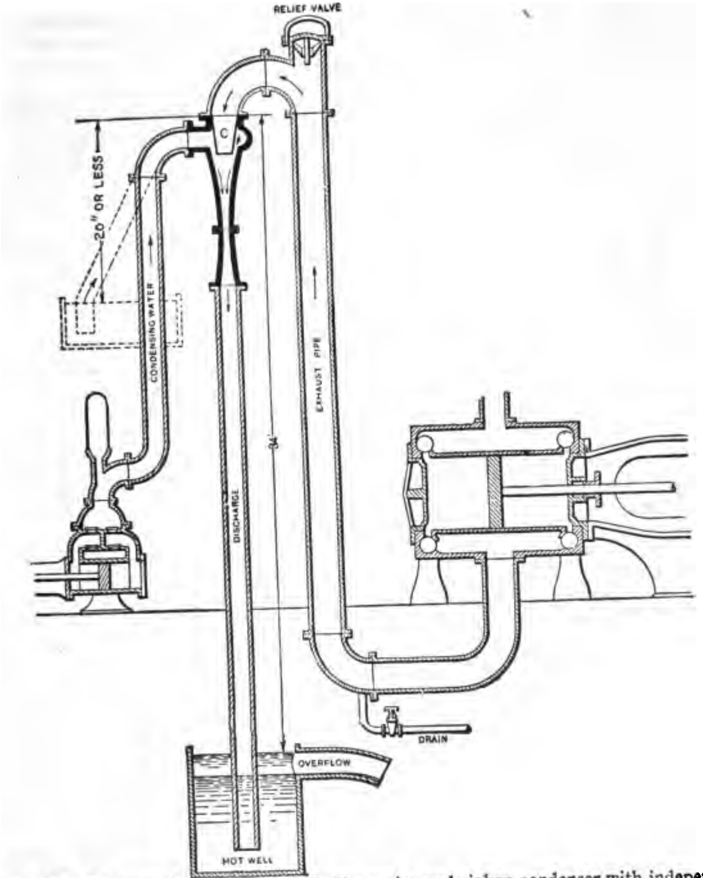


FIG. 2,213.—Diagram of connections of Corliss engine and siphon condenser with independent pump. If the level of the injection water be not more than say 20 feet before the condenser inlet, the condenser will siphon the water over as soon as a vacuum is formed in it and the pump may be dispensed with. As 20 feet is about the limit to which water may be continuously lifted by the siphoning action it follows that when the water supply is more than 20 feet below the condenser, a pump must be used as shown. This arrangement is sometimes modified by the insertion of a tank, shown in dotted lines, at about the lower limit of the siphon. This is convenient where a single action or single cylinder tank pump is used to lift the water, since such pump gives a more or less intermittent flow, whereas a practically constant flow is required by the condenser. In the tank arrangement, the pump discharges intermittently into the tank and the condenser siphons continuously from the tank.

To start: Open exhaust pipe drain, warm engine and work the water out of the cylinder; this will prime the exhaust pipe with steam and cause the relief valve to open allowing steam to escape into the atmosphere.

Since a vacuum must be formed before the condenser will siphon cooling water, open the starting or priming valve which admits water to the discharge pipe, and in falling through it, draws out the air, closing the relief valve and forming enough vacuum in the upper pipes and condenser to draw the injection water up to the condenser.

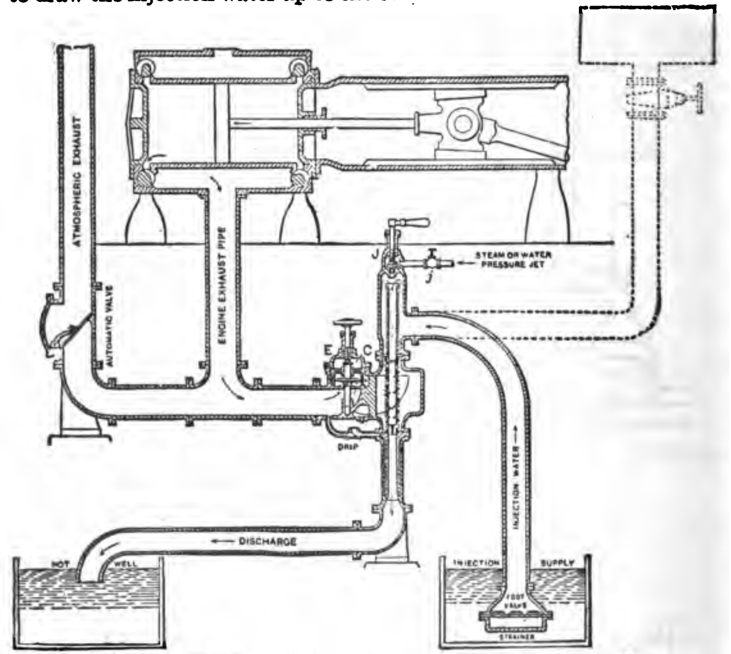


FIG. 2,214.—Diagram of Corliss cylinder and connections with Koerting exhaust steam or induction condenser. The working of the condenser is based upon the same principle as that of the steam injector. ***In operation,*** steam enters through the valve E, and passes through the inclined perforations into the central tube T, as indicated by the arrows. Owing to the velocity of its movement, the air in the condenser and the injection pipe is drawn out with it, and the atmospheric pressure on the injection supply forces the condensing water up through the pipe and into the tube T, as shown. The exhaust steam is condensed by this water and a vacuum is left in the condenser and exhaust pipe. The original velocity with which the water entered the condenser and the added velocity due to the exhaust steam enables the mingled steam and water to overcome the atmospheric pressure on the discharge end and pass out into the hot well just as the water from the injector overcomes the resistance due to friction and pressure and passes into the boiler. The velocity of the discharge is sufficient to draw out the air, cooling water and condensate, so that no air pump is necessary.

The starting valve should now be closed and the water supply adjusted by the injection valve, being guided by the vacuum gauge.

Starting with Exhaust Steam Induction Condenser.—When the condensing water is under a head, turn on the cooling water and when a vacuum is formed, start the engine.

When the cooling water must be lifted, open the steam or pressure jet valve, and as soon as this has lifted the water, start the engine.

The operation of the condenser will begin as soon as the engine exhaust reaches the condenser and when the vacuum is formed, the suction or lifting jet may be turned off.

Running.—After the engine has been started, attention should at once be given to the various oiling devices to see that they are feeding at a proper rate.

Stationary engineers, as a rule use too much oil in the cylinder, and it is safe to say that only those in charge of condensing plants where the condensate is returned to the boiler, or marine engineers, appreciate how small a quantity of oil will suffice.

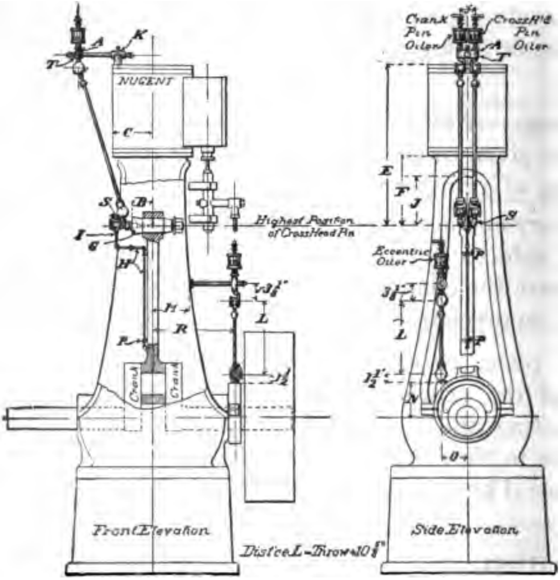
In any plant, excess cylinder lubrication should be avoided because of the expense, and in condensing plants, because of contaminating the feed water when the condensate is used. From time to time the various bearings should be felt with the hand to see if any be heating.

Lubrication.—Since a lubricant is used for removing friction where the two parts are under pressure, it is evident that under various conditions and pressures different fluids will give different results, and in order to lubricate properly it is necessary that the conditions of both the rubbing surfaces and the lubricant be considered, as well as the pressure between the moving parts.

In order to lubricate properly, the lubricant, when between the bearings, must be a fluid, enabling it to flow freely when the bearings are close fitting and under heavy pressure.

The adhesive properties, or those that cause the oil to adhere to foreign bodies, must be good so that it will stick to the metallic surfaces.

The flash point must be above any temperature that may arise in the bearing. By flash point is meant the temperature to which the oil must be heated in order that the vapors arising from it will momentarily blaze up when a flame is passed through them. If the lubricant be an oil, it must have the property of remaining fluid under low temperatures, and, if grease, it must readily liquidize with a slight rise of temperature.



FIGS. 2,215 and 2,216.—Views of a vertical engine fitted with Nugent combination telescopic oiling devices for cross head and crank pins.

Until recent years vegetable greases and oils were almost universally used for all lubricating purposes, but the methods of manufacturing mineral oils have so advanced that vegetable lubricants have been almost entirely displaced on account of their higher cost and the fact that they will not absorb oxygen from the atmosphere. There are lubricants of all kinds, at all prices and for all purposes now manufactured.

An engineer should try several of those which are claimed to be best for the work in hand and, after satisfying himself as to the most economical, use that one constantly, and for the purpose for which it is made.

One reason any lubricant fails to give satisfactory results is that it is often used for the wrong purpose. For instance, a good cylinder oil would not give satisfaction on a sewing machine. All oils are made to lubricate properly under particular conditions, and if the conditions are not right the oil is ineffective, and cannot be used economically. Another thing which governs the service rendered by an oil, and also the quantity necessary in order to obtain satisfactory results, is the bearings. They should always be in proper alignment and present a smooth, ample bearing to the revolving shaft.

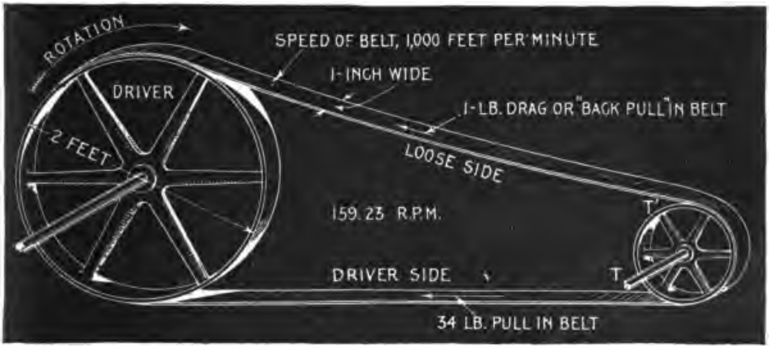


Fig. 2,217.—One horse power transmitted by belt illustrating the **rule:** ***A single belt one inch wide and travelling 1,000 feet per minute will transmit one horse power: a double belt under the same conditions will transmit two horse power.*** A pulley is driven by a belt by means of the friction between the surfaces in contact. Let T, be the tension on the driving side of the belt, and T′, the tension on the loose side; then the driving force = T − T′. In the figure T, is taken at 34 lbs. and T′, at 1 lb.; hence driving force = 34 − 1 = 33 lbs. Since the belt is traveling at a velocity of 1,000 feet per minute the power transmitted = 33 lbs. × 1,000 ft. = 33,000 ft. lbs. per minute = 1 horse power.

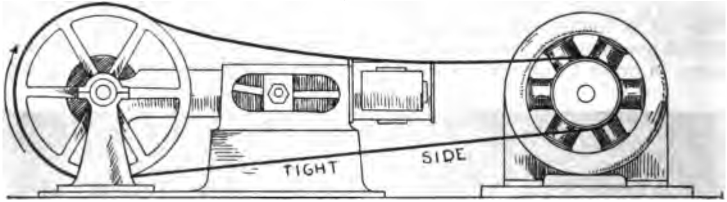
In selecting oils for various purposes the following suggestions may be of some assistance: For valves and pistons, heavy high test mineral oils with which is mixed from 3 to 5 per cent of graphite, the amount of graphite to be used depending upon the condition of the wearing surfaces, the pressure between the moving parts, and the steam pressure. The percentage is increased in proportion to the roughness of the wearing surfaces, the pressure on the bearing and steam pressure increases.

Mineral, vegetable or lard oil is suitable for ordinary machinery.

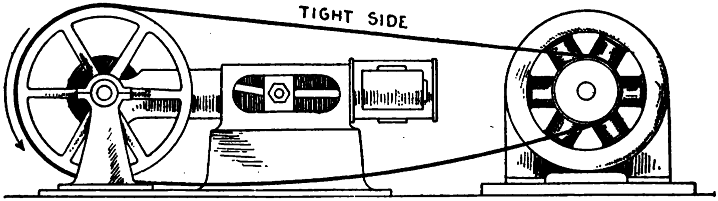
In places where the speed is low and the pressure high, lard oil, tallow or a good quality of grease may be advantageously used.

Soap with graphite is good when the surfaces in contact are wood. For high and low speed engine bearings, any heavy mineral or vegetable oil that will not deteriorate easily under ordinary temperatures is satisfactory.

When hot bearings and boxes are hard to cool and to keep cool use the same lubricant as is recommended for cylinders, and in extreme cases powdered sulphur mixed with a good mineral oil will be found effective. The



RIGHT WAY



WRONG WAY

FIGS. 2,218 and 2,219.—Right and wrong way to run a belt. The tight side should be underneath so as to increase the arc of contact and consequently the adhesion, that is to say, a *better grip* is in this way obtained.

NOTE.—There are some simple tests which may be made in determining whether or not an oil is suitable for use on machinery. Some unscrupulous manufacturers attempt to treat a cheap and inefficient oil so as to make it closely resemble high grade oil, and sell the inferior material at a slightly lower price than good oil, reaping large profits and causing the use of much larger quantities and trouble with excessive friction and gummy bearings. If a choice is to be made from the number of oils take a piece of glass and incline it. Put a drop of each kind of oil to be tested on the glass, and, after a certain length of time, several minutes, measure the distance which each drop has flowed down the glass. The one which has traveled the shortest distance will be the one which has the most body, and good body is essential in an oil for good lubrication. The second test to which oil may be subjected is removing it from the glass. This should be done after allowing the oil to stand on the glass for an hour. Those oils which contain some thick, gummy substance to make them more closely resemble high grade oil, will stick to the glass and be hard to remove. If this gummy substance is in the oil it will adhere to the bearings and cause trouble, therefore oils of this nature are to be avoided. The final test to which the usual person can subject lubricating oil is the acid test. Good oils contain no acid and will not easily become acid or rancid. Take a sample of the oil and place it in a test tube or other small vessel, and allow it to stand in the sun for an hour. Then dip into it a piece of blue litmus paper, if the paper turns red, the oil contains acid.

sulphur should be mixed with oil until the consistency is such that the oil will hardly flow.

Knocks.—The most common sign that something is loose about an engine is "knocking," as it is called. If any bearing wear perceptibly, or any fastening become loose, the engine will begin to knock. In such case, the engineer should locate the knock at once, find out the cause, and remedy same at the first opportunity. Knocks are caused by looseness, or

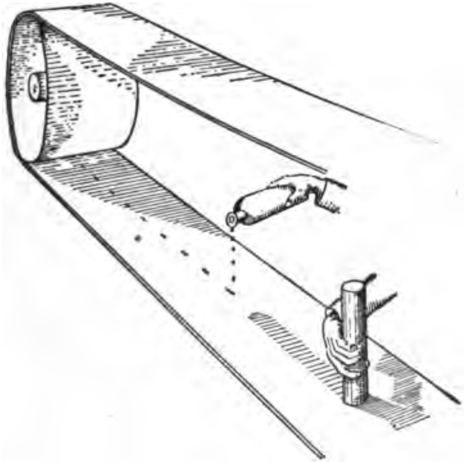


FIG. 2,220.—Method of applying dressing to a running belt. Sometimes the application of dressing to a running belt causes additional trouble. This is due to improper application. Dressing, whether liquid, paste or so called stick variety, should be applied sparingly, and at the center of the belt first, as shown, gradually working out to the edges, first applying the dressing a little to one side of the center, then a little to the other side. Treating all of one side of a belt first usually causes trouble. Such applications are efficacious in stopping slipping temporarily, while the thorough rubbing in of dressing after cleaning the surface is by all means the most satisfactory and durable treatment. Of the home made preparations probably neatsfoot oil comes first for softening very dry belts which are so hard as to begin to crack. It is not desirable to attempt to get a belt into prime condition with one treatment. This results in the use of too much oil or other dressing. It is better to treat it several times and bring it to condition gradually. For this reason all dressing should be used sparingly at each application, but applied often enough to keep the belt in the same condition after once put in order. The addition of from ½ to 1 pound of beeswax to the gallon of oil improves the latter as a dressing. First melt the wax and heat the oil, mixing the two while warm. Tallow frequently is used by first heating it until it can be rubbed into the belt. When belts run in damp basements, the addition of ½ ounce of rosin to the pound of tallow aids in eliminating the effects of moisture.

interference. Looseness in the reciprocating parts causes "lost motion" which in turn causes knocks.

In making adjustment care should be taken not to make the adjustment too tight.

The adjustment of the piston rod length may be out sufficiently as to cause the piston to hit one of the cylinder-heads. This is easily corrected by loosening the jamb nut and adjusting the threaded part of the rod. This is an example of interference.

A frequent cause of knocking is that due to lack of over travel of a reciprocating part. Thus the piston may not over travel the cylinder bore, resulting in the formation of a "shoulder" which will "interfere" and cause a knock. This is due to excess travel of the piston due to wear in the bearings.

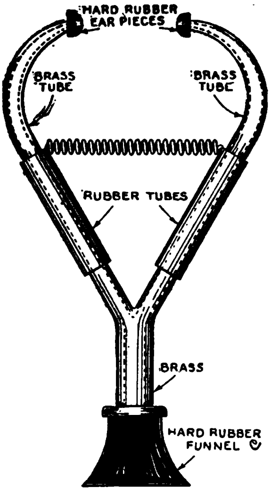


FIG. 2,221.—Stethoscope which can be used to advantage in detecting noises in engines.

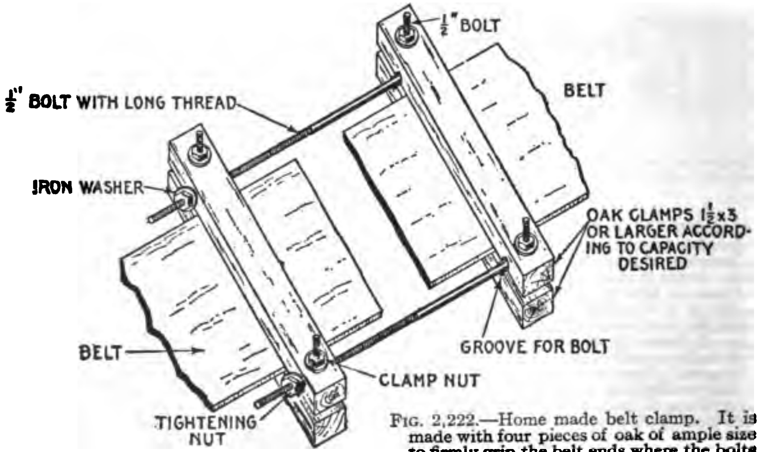
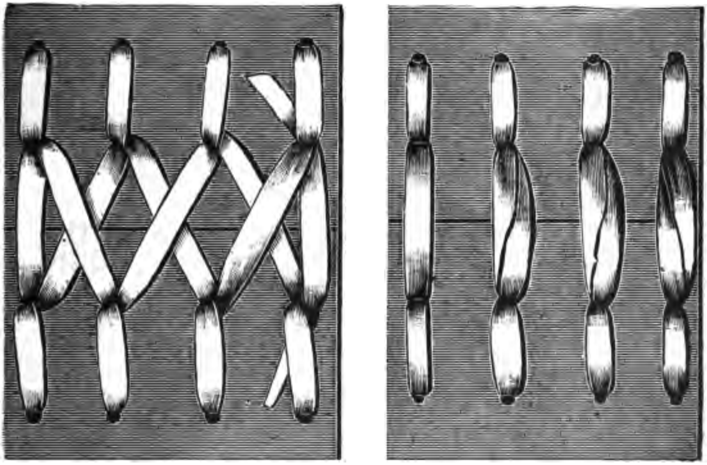


FIG. 2,222.—Home made belt clamp. It is made with four pieces of oak of ample size to firmly grip the belt ends where the bolts are tightened. The figure shows the clamp complete and in position on the belt and clearly illustrates the details of construction. In making the long bolts the thread should be cut about three-quarter length of bolt and deep enough so that the nuts will easily screw on.

On engines having stuffing boxes with screw followers, the latter is liable to come unscrewed during operation which may cause interference, especially in compactly built engines.

On some engines this applies to both the piston rod and valve stem stuffing boxes. A set screw or equivalent should be provided to lock such adjustable parts in position. Knocking may sometimes be due to improper valve setting, giving uneven distribution of the steam.

On some engines the lap of the exhaust valves must be adjusted to give enough compression to prevent knocking.



Figs. 2,223 and 2,224.—A good method of lacing a belt. The view at the left shows outer side of belt, and at the right, inner or pulley side.

In hunting for knocks it is frequently advisable to take indicator cards to see if the valve setting give the proper steam distribution.

Hot Bearings.—If a box heat in the least degree it indicates that for lack of oil or for some other reason the metals are wearing together. The first attention is to provide plenty of oil. On some sight feed oil cups, provision is made for such emergency by arranging them so that the needle valve may be opened wide by turning the lever midway between the on and off positions.

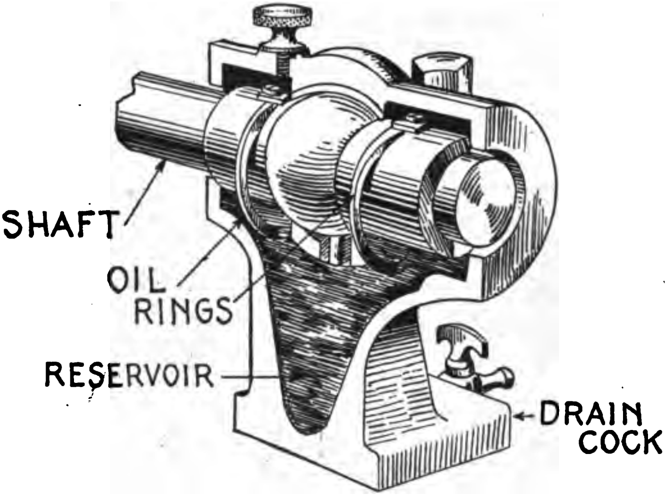
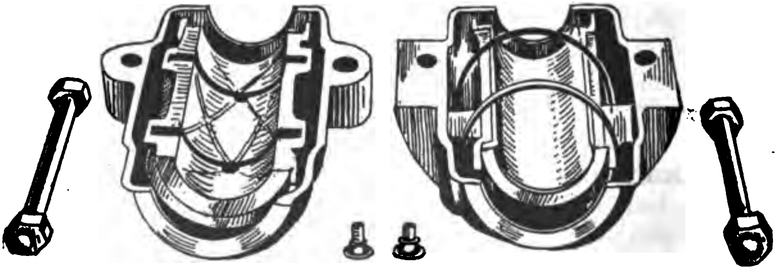


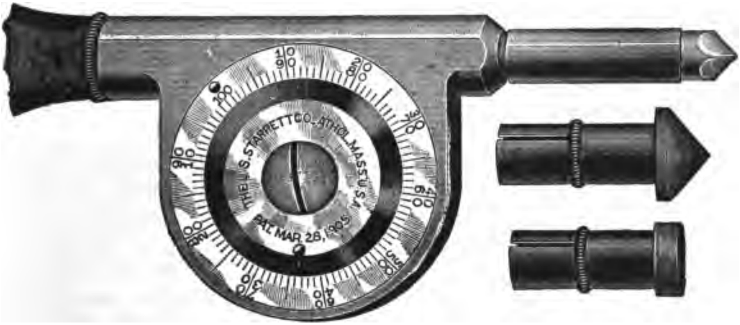
FIG. 2,225.—Sectional view showing a ring oiler or self oiling bearing. As shown the pedestal or bearing standard is cored out to form a reservoir for the oil. The rings are in rolling contact with the shaft, and dip at their lower part into the oil. In operation, oil is brought up by the rings which revolve because of the frictional contacts with the shaft. The oil is in this way brought up to the top of the bearing and distributed along the shaft gradually descending by gravity to the reservoir, being thus used over and over. A drain cock is provided in the base so that the oil may be periodically removed from the reservoir and strained to remove the accumulation of foreign matter. This should be frequently done to minimize the wear of the bearing.



FIGS. 2,226 to 2,231.—Self oiling, self aligning bearing open. Views showing oil grooves, rings, bolts, etc.

If flooding do no good, take the bearing apart and clean it thoroughly, then apply a coat of white lead mixed with good oil, being careful not to get any dirt or grit on the bearing or in the part; assemble bearing and do not adjust any closer than necessary.

The Feed Pump.—On some engines the feed pump is connected direct to the engine. This arrangement, especially in the case of high economy engines is very desirable because it is the most economical method of feeding the boiler, and also very



FIGS. 2,232 to 2,234.—Starrett's improved speed indicator. *In construction,* the working parts are enclosed like a watch. The graduations show every revolution, and with two rows of figures read both right and left as the shaft may run. While looking at the watch, each hundred revolutions may be counted by allowing the oval headed pin on the revolving disc to pass under the thumb as the instrument is pressed to its work. A late improvement in this indicator consists in the rotating disc, which, being carried by friction, may be moved to the starting point where the raised knobs coincide. When the spindle is placed in connection with the revolving shaft, pressing the raised knob with the thumb will prevent the disc rotating, while the hand of the watch gets to the right position to take the time. By releasing the pressure the disc is liberated for counting the revolutions of the shaft when every 100 may be noted by feeling the knob pass under the thumb lightly pressed against it, thus relieving the eye, which has only to look at the watch to note the time.

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*NOTE.—Independent (steam driven) boiler feed pumps are extremely inefficient, consuming in some cases as much as 200 lbs. of steam per horse power per hour, as compared with 15 lbs. or less for a triple expansion marine engine. The wastefulness of an independent pump may be reduced by passing the exhaust through a secondary feed water heater, or connecting it with the receiver in the case of a compound engine, thus virtually converting it into a compound pump.

little space is required and considerable piping avoided that would be necessary with an independent pump.*

Stopping.— In order to stop a simple non-condensing engine, all that is necessary is to slowly close the throttle valve, allowing the engine to come gradually to rest. However, in some condensing arrangements the process is not so simple and unless properly performed is liable to be attended with serious consequences.

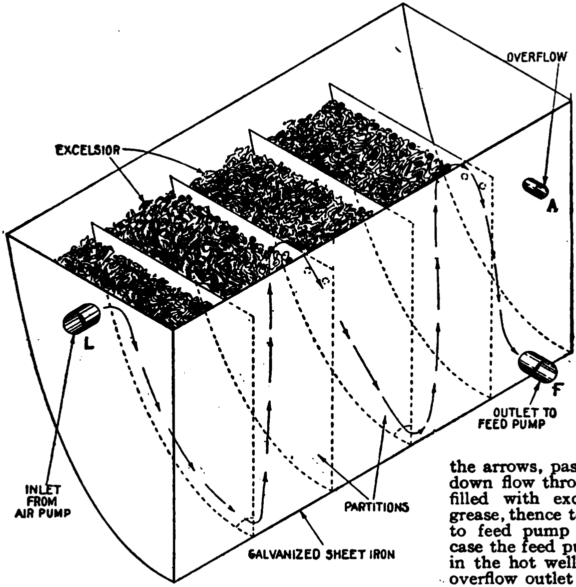

FIG. 2,235.—***Hot well** of steamer "**Stornoway**."* In operation, condensate from the air pump enters at **L**, and as indicated by the arrows, passes by alternate up and down flow through four compartments filled with excelsior to abstract the grease, thence to last compartment and to feed pump through outlet **F**. In case the feed pump fail, water will rise in the hot well and pass out through overflow outlet **A**, which connects with the tank. This is an efficient and easily constructed hot well. It is made of sheet metal either copper or galvanized iron.

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*NOTE.—Usually the feed pump is attended to by the fireman, but when it is direct connected to the engine, being in the engine room, it naturally receives attention from the engineer. After starting, the engineer should see that the feed pump is properly working. This should be done at once, especially when water tube boilers are used. The operation *of the feed pump* is indicated: 1, by condition of the hot well; 2, by temperature of the feed pipe line, and 3, sometimes by pounding of the check valves. If 1, the hot well remain flooded; 2, the pump and pipe line feel hot, and 3, no sound be heard at the check valve, the pump is not working. The suction line from the pump has one branch to the hot well and another to the tank with a valve on each. In such arrangement, the valve on the tank branch, called the "make up" valve is kept closed until the water level in the boiler gets below the working point. The water level may be quickly brought back to the desired point by opening the make up valve and closing the hot well valve. Usually it is only necessary to "crack" the make up valve to maintain a central level. Since the capacity of the pump must be in excess of running requirements, when pumping from the hot well a considerable amount of air is delivered into the boiler. To correct this, tap the feed line between pump and boiler, run a line to hot well and attach a valve to same. With this arrangement the make up valve can be left open and the excess allowed to return to the hot well by adjusting the valve.

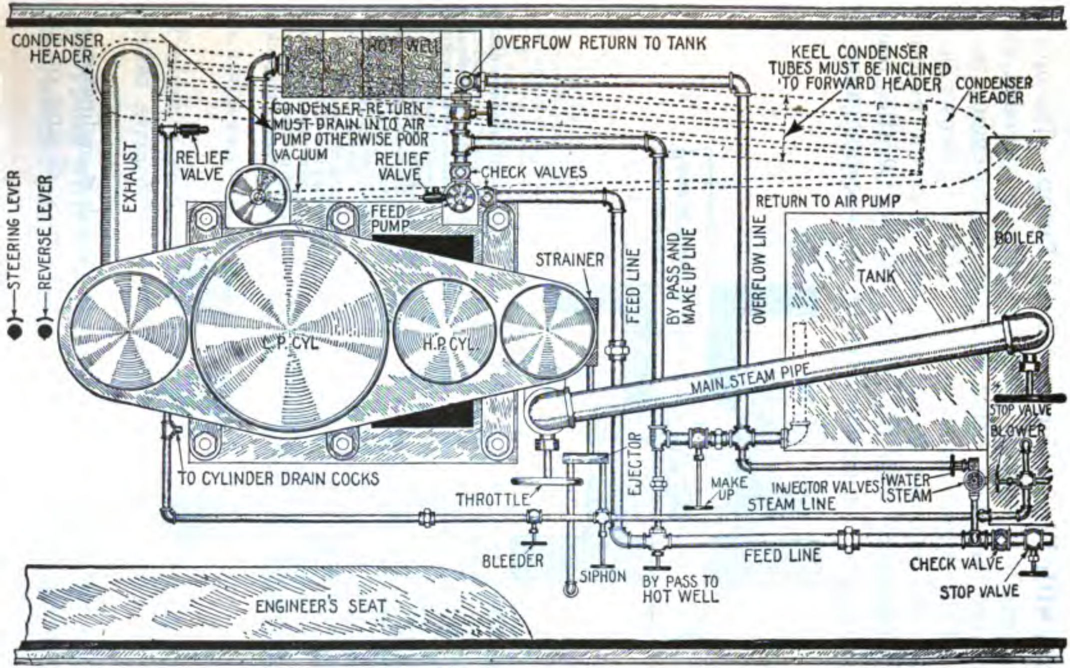


FIG. 2,236.—General arrangement of machinery and piping of author's steamer ***"Stornoway,"*** showing placement of the various valves, etc., for one man control. ***Note position:*** 1, of throttle, reverse and steering levers; 2, by pass, make up, siphon, and bleeder valve, all within reach from engineer's seat; 3, of tank under boiler where water absorbs some heat from ash pan. ***With keel condenser,*** the tubes and return pipe ***must*** be inclined so *condensate* ***will drain*** *from condenser* ***inlet to air pump,*** *otherwise* ***little or no vacuum.*** Relief valves should be placed on exhaust and feed lines. ***In piping,*** a liberal number of unions should be used and special fittings avoided. There should be valves next to boiler on feed and main steam lines. The injector should receive its steam from separate connection to boiler instead of from main steam pipe.

Stopping with Direct Connected Air Pump.—The operation of stopping with a direct connected pump is attended with no danger since the pump, being attached to the engine, is in motion until the engine comes to rest. If the condenser be of the jet type the engineer should gradually close the injection valve simultaneously with the closing of the throttle, being guided in closing the injection valve by the vacuum gauge; if the vacuum fall before the throttle is closed it indicates insufficient cooling water, caused by a too quick closing of the injection valve.

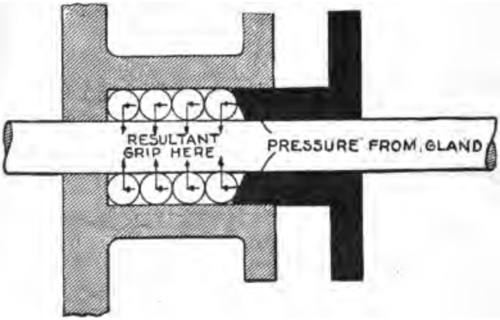
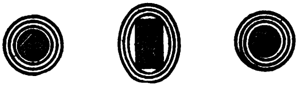


FIG. 2,237.—Principle of "Moncky wrench" packing. Packings of this type include the ordinary square flax, round core and gum core packings and what is commonly known as "red core spiral." The round and flat gum core packings are made by braiding successive sheaths about a flat or round gum core. These packings form about the cheapest steam packings that are on the market. They depend entirely for their success on a pressure from the gland and for that reason they are classed as "Moncky wrench" packing. As the pressure from the gland is generally much stronger than necessary, there is produced on the rod an unnecessary friction, and as this friction is on a moving rod there is done at every stroke a certain amount of work that is unnecessary, which work is required simply to pull the rod through the packing. This, of course, requires steam and steam requires coal and there is actually consumed a certain percentage of the work from the engine in overcoming the excess friction caused by the packing. Sometimes this packing is made by wrapping a piece of duck around a gum core, but the principle of application is the same and from an economical standpoint this class of packing, which was the class first brought out, is really the poorest, the friction almost invariably being excessive and the consumption of the steam on account of the packing very large.



FIGS. 2,238 to 2,240.—Typical cross sections of Moncky wrench packings.

Stopping with Jet Condenser.—When shutting down an engine with an independent jet condensing apparatus, *close the engine throttle first, and when the engine has stopped,* ***and not until then,*** *close the injection valve, and lastly shut down the air pump.*

By shutting off the water supply before the air pump is stopped, the water already in the condenser and pipes is pumped entirely out and there is no danger of it getting into the engine cylinder.

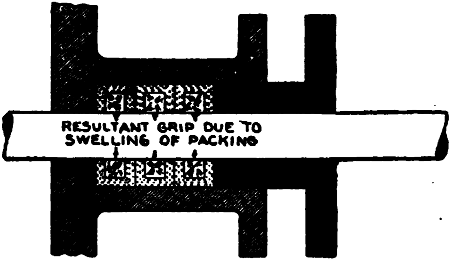
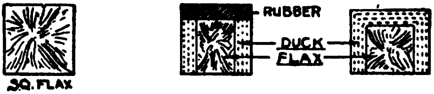


FIG. 2.241.—Principle of moisture packing. Flax packing might be classed under this head. These are packings that require moisture to expand them. The best example is the case of flax and duck, sometimes made up with a rubber back, sometimes with the duck passing completely around the back of the packing. In this case the flax swells up with the moisture and does the packing, while the duck stands the wear and tear. This is more satisfactory than the first class. It has been used for many years by old engineers. It is not a packing, however, that would be available where the steam was entirely dry when it reached the box, or where it was superheated. Of course steam that is dry when it reaches the cylinder is not dry when it reaches the box, or when it enters the box, for the reason that the box is more exposed and consequently is much cooler and the rod is one-half the time out in the cool air, and since as high a temperature as the live steam cannot well be maintained, the steam in the box will be moist. If the steam be originally superheated, however, it may get in the box as dry steam, in which case it is very hard on a packing that requires moisture to cause it to swell and thus produce the requisite grip on the rod.



FIGS. 2.242 to 2.244.—Typical cross section of moisture packing.

Failure to properly follow the above directions may result in a wrecked engine.

Stopping with Siphon Condenser.—The characteristics of this condenser being the inability of the cooling water to reach the cylinder, stopping is not accompanied with any danger as with a jet condenser.

To stop the engine, *gradually close the throttle valve and after the engine has come to rest shut off the cooling water and open the exhaust pipe drain valve.*

Stopping with Exhaust Steam Induction Condenser.—*First stop the engine, when the operation of the condenser will cease if the cooling water be under a suction lift;* if it be under head, stop the engine first and then shut the valve in the water supply pipe.

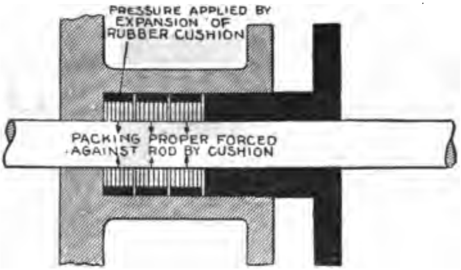
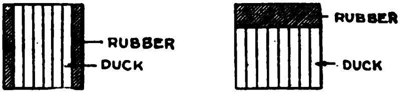

PRESSURE APPLIED BY EXPANSION OF RUBBER CUSHION
PACKING PROPER FORCED AGAINST ROD BY CUSHION

FIG. 2,245.—Principle of expansion packing. This packing expands but not from the pressure from the gland. This is generally caused by the expanding of some material used as a cushion, under the influence of heat. The material almost universally used is rubber. Rubber when heated will expand, and if it be used in connection with the packing, it will when heated force the packing against the rod and give an easy cushion effect on the rod, which is entirely independent of the gland. Since this is true, there is no reason for the engineer to use a Moncky wrench, and consequently there is no opportunity to force the packing too tightly against the rod and produce undue friction. It is therefore a packing that is more economical so far as consumption of steam is concerned, since there is a lighter load on the engine due to the decrease in the friction of the packing on the rod. An expansion packing is a better packing than that which depends on pressure from the gland, such as a Moncky wrench packing, being not only more economical of steam, but also easier on the rod. This same result is produced in certain metallic packings by the use of springs.



FIGS. 2,246 and 2,247.—Typical cross sections of expansion packing.

2. CARE

The term care is here used in the sense of maintenance, that is, in the constant running of an engine certain supplies such as oils, packings, gaskets, etc., must be adjusted and renewed from time to time as required.

The subject of lubrication has been mentioned under "running," and treated at length in preceding chapters, hence no further remarks are necessary.

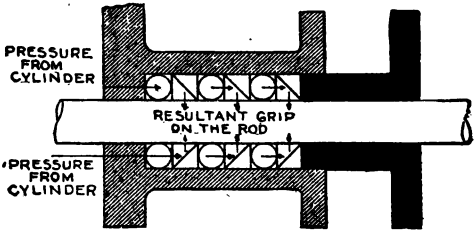
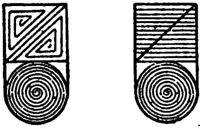


FIG. 2,248.—Principle of automatic diagonal packing. This packing does not depend on pressure from the gland, or on any expansion due to heat or due to the swelling of any substance, but does depend on pressure from the cylinder. This packing is an automatic packing commonly called "diagonal" packing, because it contains wedges which are formed by cutting a square section diagonally. There are a number of diagonal packings, differing from each other in the form and shape of wedges, but all of them depend upon the same fundamental principle that the pressure from within the cylinder will force the wedge next to the rod in toward the rod, and, of course, that means that the grip which it has on the rod will vary with the pressure forcing it, in other words, with the cylinder pressure.



FIGS. 2,249 and 2,250.—Typical cross sections of diagonal packing.

Packing.—The object of a packing is to prevent a leak of any fluid, which may be either a liquid or a gas. This may be a comparatively simple case as when gaskets are used to prevent steam escaping at a joint, or it may be quite complicated, as by the use of packing on piston rods.

Packing is held between the faces of joint by the friction produced by the clamping pressure of the two parts, one against the other; when held this way, as for instance, between the end of a cylinder and the cylinder head it is called a gasket.

With high pressure this friction may not be sufficient to resist the internal pressure exercised against the thin edge of the packing. Retaining rims, inside or outside, around the opening, or male and female projections on the face of the flange, are, therefore, provided with high pressure, to relieve and remove the packing from the influence of the inside pressure.

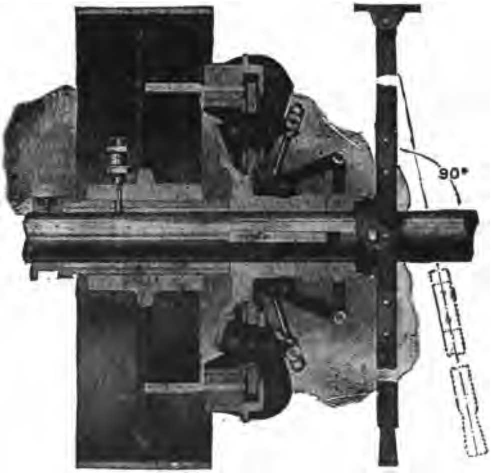


FIG. 2,251.—The Hill friction clutch pulley for power control. The clutch mechanism will start a load equivalent to the double belt capacity of the pulley to which the clutch is attached.

The packing is materially affected by the retained fluid, according to its temperature, density and chemical reaction.

The packings may be classified from this point of view into packings for

1. Steam;
2. Water;
3. Air, ammonia and oil.

With respect to the nature of the joint, they may be classified as those intended for

1. Stationary parts, or
2. Moving parts.

Steam Packings.—These packings, especially for high pressures and temperatures, should be as nearly incombustible as possible; they should not be decomposed or easily burned on to the faces of the joint.

On this account metallic packings are chiefly used for this purpose, and next in desirability are the fibrous packings, made from mineral substances like asbestos.

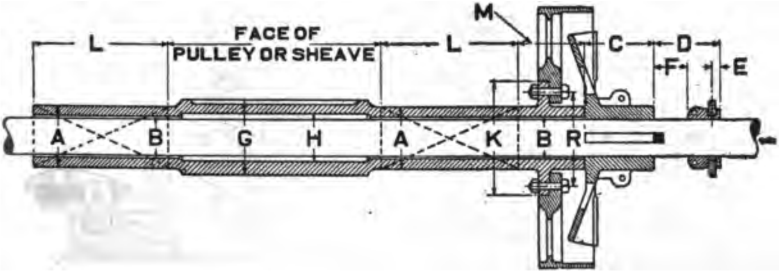


Fig. 2,252.—Quill drive. This is the proper transmission arrangement substitute for heavy service, requiring large pulleys, sheaves, gears, etc. It is a hollow shaft supported by independent bearings. The main driving shaft running through the quill is thus relieved of all transverse stresses. The power is transmitted to the quill by means of a friction or jaw clutch. When the clutch is thrown out, the pulley or sheave stands idle and the driving shaft revolves freely within the quill. As there is no contact between moving parts there is no wear. Jaw clutches should be used for drives demanding positive angular displacement. They can only be thrown in and out of engagement when at rest. All very large clutch pulleys, sheaves, or gears designed to run loose on the line shaft are preferably mounted on quills. The letters A, B, C, etc., indicate the dimensions to be specified in ordering a quill.

For low pressure or exhaust steam, other fibrous, or rubber, packings are cautiously used with advantage by engineers.

Around condensers and air pumps the packings should possess durability against the effects of mineral oil in steam or water.

Water Packings.—For this service, packings may be either metallic or fibrous. They should, however, be of such material as to have no galvanic reaction upon the material of the pipes or machines, as sometimes experienced with copper rings.

They should not be decomposed or rotted by soaking, but should retain their elasticity, or softness, even under frequent making or breaking of the joint.

Air, Ammonia, and Oil Packings.—The requirements for these purposes are limited, and the regular steam and water packings generally can be applied to these needs. The air pipes of forced ash pit, or induced draft, systems require some packing of fibrous, or rubber, material for cold air piping, and of asbestos or metallic material, for hot air piping or ducts.

Ammonia piping may have metallic, rubber or fibrous packing, according to the judgment of the engineer.

The main requirement in packing for oil piping is that it should not be dissolved, or decomposed, by oil, as are some of the soft rubber materials used for packing.

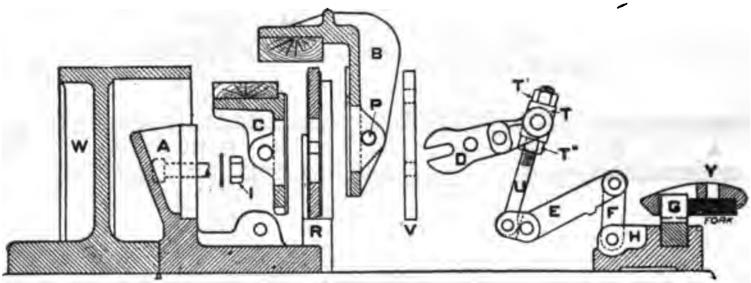


Fig. 2,253.—Sectional view of Hill clutch mechanism. In every case the mechanism hub A, and in a clutch coupling the ring W, is permanently and rigidly secured to the shaft and need not be disturbed when removing the wearing parts. When erected, the adjustment should be verified, and always with the clutch and ring engaged and at rest. If the jaws do not press equally on the ring, or if the pressure required on the cone be abnormal, loosen the upper adjusting nuts T', on eye bolts and set up the lower adjusting nuts T", until each set of jaws is under the same pressure. Should the clutch then slip when started it is evident that the jaw pressure is insufficient and a further adjustment will be necessary. All clutches are equipped throughout with split lock washers. Vibration or shock will not loosen the nuts if properly set up. The jaws can be removed parallel to the shaft as follows: Remove the gibs V, and withdraw the jaw pins P, then pull out the levers D. Do not disturb the eye bolt nuts T' and T". The outside jaws B, can now be taken out. Remove the bolt nuts I, allowing the fulcrum plates R, to be taken off. On the separable hub pattern the clamping bolts must be taken out before fulcrum plate is removed. The inside jaws C, may now be withdrawn. Oil the moving parts of the clutch. Keep it clean. Examine at regular intervals.

Packings for Stationary Parts.—For these parts packings are generally applied in the forms of sheets, called gaskets, thin and well spread out between flat faces or flanges of valves, cylinders, valve chests, etc. Occasionally engineers prefer to use a round metal, or fibrous, ring in a recess, instead of a flat sheet, as it is more easily made and kept tight, while bearing only on a narrow line, or ridge, all around the opening.

The choice of the packing is greatly influenced by the consideration of whether the joint is to be broken often, or will remain comparatively undisturbed for a longer period.

In the latter case materials may be employed that cannot be used over again, such as red lead or iron rust cement, with or without fibrous gaskets, while, in the first case, quickness of making and preservation of the packing for continued use will be of importance.

The packings used for stationary parts may be divided into two classes: those that cannot be used again, such as iron rust cement, red lead, with or without chopped hemp, and graphite; and those that can be more or less frequently used again, such as the metallic gaskets, asbestos, pure rubber, mixed rubber and canvas, canvas or paper soaked in linseed oil, or applied in connection with red lead or graphite.

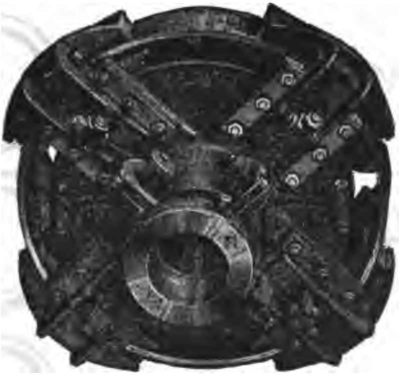


Fig. 2,254.—Hill clutch mechanism Smith type. The friction surfaces are wood to iron, the wood shoes being made from maple. All parts of the toggle gear are of steel and forgings with the exception of the connection lever which is of cast iron.

Packings for Movable Parts.—These are generally arranged in coils of several thicknesses to gain depth in the direction of the motion, or in the longitudinal axis, like the coils in the stuffing boxes of piston rods, valve stems, condenser tubes, or in the packing of air pump pistons.

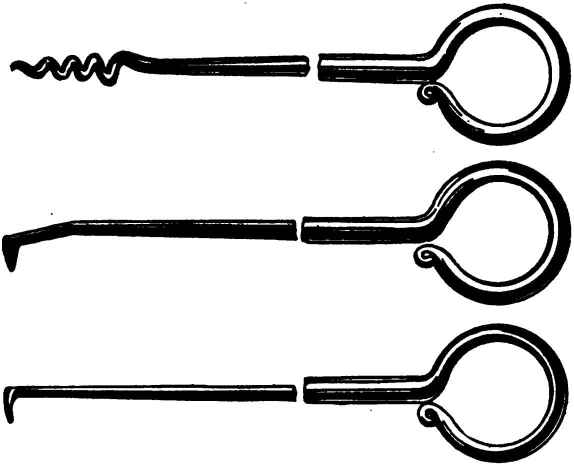
Packings for movable parts should be employed in good depth, with relatively slight pressure upon them, to retain elasticity and secure good durability and long life with little attendance and adjustment.

The renewing of stuffing box or piston packing is generally a troublesome experience, attended with great expenditure of time and labor, like that of the condenser tubes, where the water chests must be removed, in order to get at the stuffing boxes, or that of the air pump piston, where the top covers, valves and moving parts must be removed for access to the interior.

Oils, red leads and other substances that harden or bake should not be applied to packing for movable parts; also the packings themselves should be of substances that do not burn. Softness, pliability and elasticity should be their chief properties, and these should be retained as long as possible.

The packing used for moving parts is generally employed in round or square rings or strands of soft metal, of asbestos, of rubber, of combinations, or of hemp made up into braids.

The packings, particularly the soft ones, are sometimes soaked with tallow, or graphite powder is rubbed into them to increase their durability and resistance to wear.



FIGS. 2,255 to 2,257.—Engineer's packing tools for use in removing and inserting packing.

Methods of Applying Packings.—The way in which a packing is applied varies greatly according to material, position, finish of faces of joint, nature and duty of part in need of packing. It is expedient to apply the hand made packings in a heavier layer than the nearer uniform commercial packings.

On rough surfaces more packing material is needed than on smooth or finished surfaces. Stationary parts, with a higher clamping pressure, need much less packing than movable parts, where the packing is compressed comparatively lightly.

The pressure upon the packing is generally exerted by bolts, nuts, or tap screws.

For stationary parts the packing is clamped between stiff flanges which with the numerous bolts, exert a very heavy pressure upon the packing. The bolts should be spaced closer together with light flanges than with heavy flanges, as otherwise a slight bulging of the flange between the bolts may take place, leading to leakage and final tearing and blowing-out of the packing material.

For moving parts the packing is forced into a special recess of the stuffing box or piston by a gland or follower ring. As the danger of blowing out is practically removed in these places, the area under pressure being generally much smaller than with stationary parts, and, as the desired elasticity of the packing makes a low clamping pressure advisable, fewer bolts are required.

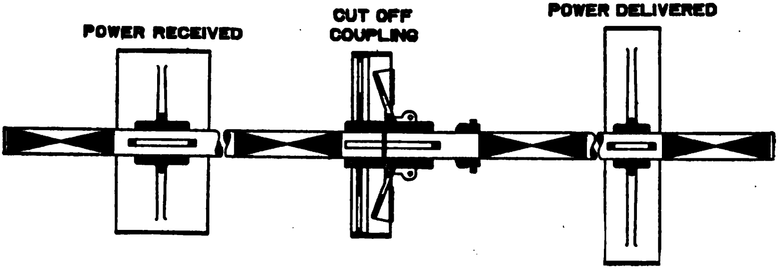


Fig. 2,258.—Cut off coupling for power transmission by line shafting. It is used to cut off a driving shaft from a driven shaft. Its use obviates the use of a *quill*, such as is shown in fig. 2,858.

Iron rust cement is applied to cracks, patches and repairs of permanent character in general, where the interstices must be filled by a body that becomes firmly solid and iron like in nature. It is made up from fine iron filings into a cement or paste by treatment with acid. This turns the metallic iron into iron rust, baking the particles together.

Red lead is made up from the powder by mixing thoroughly with linseed oil, until it is of the consistency of paste.

It is used for the largest number of joints not too frequently broken, where the hardening of the red lead by the drying of the oil is of advantage for permanent tightness in the joints.

A thin wash of red lead, or only linseed oil, in connection with canvas or paper, is used with advantage for a large number of machinery joints, like cylinder covers, valve chest covers, etc., which require comparatively frequent lifting.

Graphite is applied, in conjunction with metallic, asbestos, rubber and fibrous packings, as powder or made up with a little oil or tallow to a paste. It is excellent for all joints that are frequently broken; prevents burning of the packing to the faces, and effects a saving of material by keeping the gasket in good condition for renewed employment.

Canvas, Paper, and Hemp Packings.—These are employed generally only temporarily, where lack of the higher grade obliges the engineer to use a substitute. They are used soaked in oil and coated with graphite, grease or red lead in order to give more body and tightness.

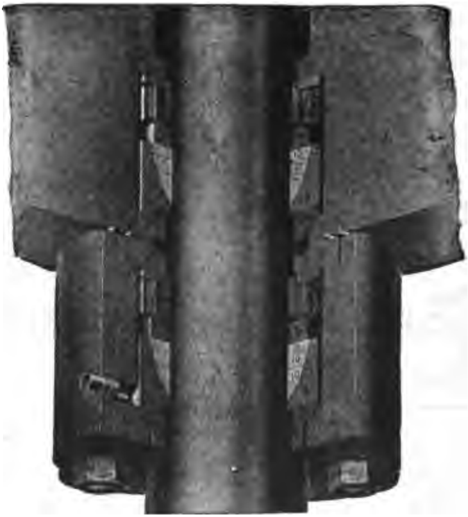


FIG. 2,259.—U. S. Class 1, metallic double packing. The inner or upper set of Babbitt metal rings does most of the work. It is separated from the rest of the packing by a dividing piece, 8; outside or below this comes the second set of rings. Should any water or steam leak through the first set of rings it is caught by the second set and can be drained off; also should the first set of rings wear out, the second set is already on the rod to take up the work of the first set. ***The principle*** of this packing is that *the soft metal rings* 10, *are forced by the steam pressure into the vibrating cup* 6, *and against the rod or stem.* Flexibility is attained by the combination of the ball joint 7, with the sliding face of the vibrating cup 6. This enables the packing to accommodate itself to rods out of alignment. ***The parts are:*** 1, preventive; 2, spring guards; 3, spring bushings; 4, springs; 5, followers; 6, vibrating cups; 7, ball joints; 8, ball seat; 9, gland; 10, Babbitt metal rings.

Hemp packing was formerly made by the engineers from the raw material into braids of varying thickness, shapes and length. The modern commercial packings, however, made up and ready for use, save so much time and labor, and prove in the end so much cheaper, that the practice has practically died out, being employed only in emergencies and special cases.

Asbestos Packing.—This is made in shape of sheets, round or square strands, wicks, etc., from the fibrous raw mineral material. It has the valuable quality of being incombustible and imperishable, and is thus eminently adapted to all high pressure pipe and machinery joints around boilers and engines. It ranks with metallic packings in this particular. One great advantage is its softness, fibrous nature and comparative elasticity, while it is, on the other hand, not nearly as durable as metallic packing, and, therefore, more expensive. Some engineers employ asbestos packing, together with metallic packing, in stuffing boxes, thus combining the softness and elasticity of the asbestos with durability of the metallic packing.

Rubber Packing.—This is the most frequently employed material. Its great elasticity, comparative strength and toughness, tightness and moderate price make it an excellent packing for all water service, joints around pumps, pipes, manifolds and valves. Numerous engineers prefer some one of the particular makes of this packing for low pressure and exhaust steam joints.

Rubber packing is made in sheets of all sizes, gaskets, round and square strands, washers, etc., generally with a body of canvas in one or more layers, that increase the strength and resistance against blowing out.

Graphite, rubbed on the faces of rubber packings, effectively prevents burning on, thus allowing the gaskets to be used over.

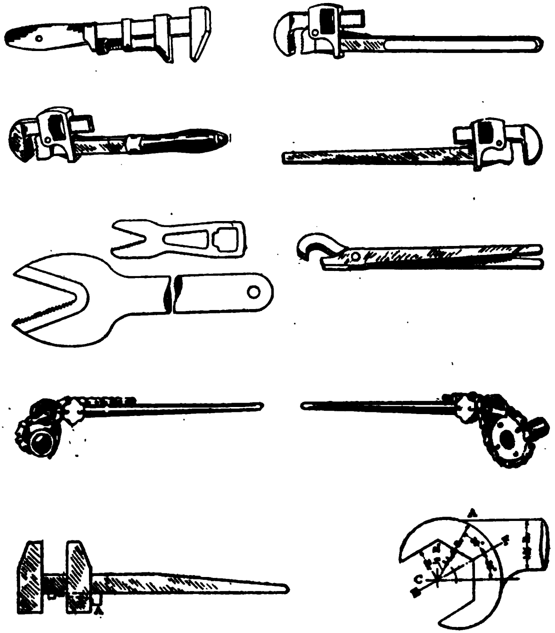
Metallic Packing.—Usually metallic packing is made from the soft metals, which, under pressure, yield and shape themselves to the bearing surface. It is almost universally employed, and proves excellent for high steam pressure, where fibrous packing would burn and last but a very short time. The elasticity of metallic packing is, however, considerably less than that of the fibrous packing, and this is a certain disadvantage, particularly in stuffing boxes. To overcome this lack of elasticity, certain constructions of metallic stuffing boxes employ springs, which tend to prevent gripping of the packing on the rods. The greater complication of this arrangement is objected to by many engineers, and the simple, more rigid arrangement preferred.

Copper, lead, Babbitt metal and similar compositions of soft nature are employed with advantage in form of rings, sheets, stuffing box coils, etc. Copper is employed in rings of round, or triangular, sections, or in corrugated discs for packing on flanged boiler valves, manholes, pipe flanges or engine flanges. It is very durable and lasting, but, with iron flanges or faces, it sets up galvanic action in some liquids, thus quickly destroying the smoothness of the faces.

The lead or Babbitt metallic packings are used in sheets for flanges, or in split rings for stuffing boxes. Their melting point should be considered for high pressure steam. Babbitt metal may be mixed so as to resist very effectively any temperatures in use.

3. REPAIR

A thorough overhauling of an engine is occasionally required in order that general repairs can be made. The principal reason for taking an engine apart is to ascertain the exact condition of the pistons and bearings, and cylinder walls.



Figs. 2,260 to 2,270.—Various wrenches. Fig. 2,260 Moncky wrench (erroneously spelled "monkey"); figs. 2,261 to 2,263, Stillson pipe wrenches; figs. 2,264 and 2,265, alligator pipe wrenches; fig. 2,266, pipe tongs; figs. 2,267 and 2,268, chain wrench showing application on pipe and on flange; fig. 2,269, type of key wrench; fig. 2,270, offset wrench with 30° offset.

NOTE.—The so called "monkey" wrench was invented by Charles Moncky, who sold his patent for a comparatively small sum, and who lived during the later years of his life in the Williamsburg section of Brooklyn, N. Y. The author refuses to dub such a valuable invention with the title of monkey and accordingly adheres to the correct spelling, capital and all: "Moncky."

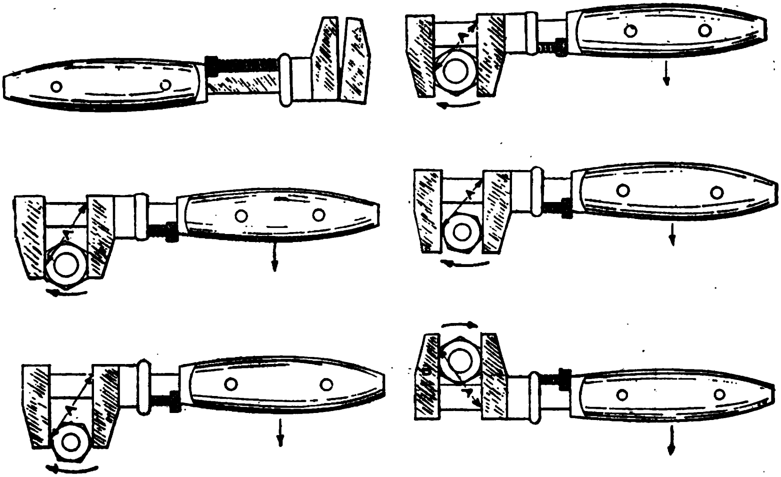
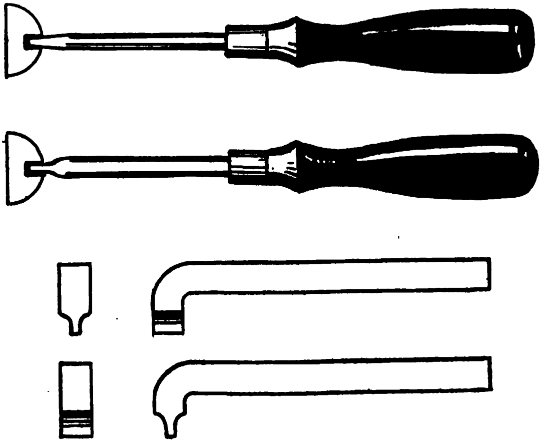


FIG. 2,271 to 2,276.—Use and abuse of the Moncky wrench. There is probably no wrench in existence more misused than this type. They are designed for use by hand on the nuts that they will take, and will stand all work put on them *with the hands only*, giving good service if they are always applied in the proper manner. Invariably on calling for a Moncky wrench, one is brought out looking very much like fig. 2,271, with the jaws at an angle with each other when closed, instead of being parallel, as they should be. This condition is caused by abuse in the use of the wrench, principally through ignorance. When the Moncky wrench is applied as shown in fig. 2,272, there will be no difficulty about its doing its work. When the nut is to be turned in the direction shown by the arrow, the wrench must always be' applied as shown. Not only must the wrench be applied in the right direction, but it must come down full on the nut as far as it will go, the reason being that the force which tends to break the wrench or bend the jaws into the shape of fig. 2,272 is along the line A, fig. 2,272, and with the wrench clear down the leverage is reduced to a minimum. In fig. 2,273 it will be seen that the line A, is increased by not letting the wrench down on the nut, although the jaws are closed up tight on the nut. Fig. 2,274 shows line A, not greatly increased, but through the loose adjustment of the jaws the corners of the nut get a greater purchase on the wrench and tend to push the jaws apart more forcibly. In figs. 2,275 the two forces which tend to ruin the wrench have the best opportunity on account of the poor adjustment of the width between the jaws and the wrench resting high up on the nut. These are common faults in the use of Moncky wrenches, but the abuse most common is illustrated in fig. 2,276 which shows the wrench upside down. As soon as the force is applied in the direction of the arrow, the outside jaw takes hold of the nut at B, and line A is increased at once. This is positively a case where there is only one way that is right, and any other is wrong. Not only can wrenches be saved by applying them as in fig. 2,272, but many skinned knuckles and mashed fingers might have been prevented, and of more importance, considerable time saved. When a Moncky wrench cannot be applied to its work properly, some other type of wrench should be used. Another infallible rule for the right use of a Moncky wrench, is to never use a piece of pipe over the handle to increase the leverage. Nor is it right to strike on one of these wrenches with a hammer. Most of the ruined handles on Moncky wrenches come from these two sources. All types of wrenches should be used with care and precision, and should always be placed squarely on the nut and made to fit it snugly. With socket wrenches it is often impossible to use them unless they are held squarely and snugly to the work.

Each part as it is removed should be cleaned. As soon as one part is unjointed or uncoupled, insert its pins or screws in their proper place before laying aside. This will prevent any small parts being misplaced.

In disassembling, the cylinder is removed, after the engine has cooled sufficiently, and the bore of the cylinder inspected for smoothness and wear. The piston follower ring is taken off, and the condition of the packing ring and the springs that force it out against the walls of the cylinder is ascertained. The nut that holds the piston on the rod is inspected, to ascertain whether it has remained screwed down hard and is securely locked in place.



Figs. 2,277 to 2,282.—Forms of screw driver. A screw driver with a wedge shaped head which fits the slot of the screw, as fig. 2,277, should not be used, and yet is universally sold by manufacturers and used in that form. It is plain that this form of screw driver never fits the slot in the screw head and takes as much force to hold the driver in place as it does to drive the screw. Another fault is that it puts a strain on the screw head where the power which tends to break it apart is greatest. When the head of the screw driver is ground so that it takes hold of the screw head in the bottom of the slot, as in fig. 2,778, the strain on the screw head is at a minimum, and the power of the operator is all spent in driving the screw alone. In some cases it is impossible to use ordinary screw drivers, owing to the cramped space, and the driving force must be applied at right angles to the driving line. Figs. 2,279 to 2,282 show views of two screw drivers which are useful in such cases. It can be seen that with the first type A, shown the screw head can be moved one-quarter of a turn and be picked up with the second type B, and turned another quarter, the two used alternately until the work is done.

Broken or worn bolts and nuts of the follower ring are replaced, and the springs are reset, so that a uniform pressure is exerted on the packing ring.

If liners be fitted they are inspected for preservation of the packing at the outer end, while the tap screws at the inner end are carefully examined to see if they be loose or broken.

For large engines or tandem arrangements manholes are fitted that allow access to either end of the cylinder, without removing the piston.

The valves are removed and the seats inspected for smoothness or wear.

Solid piston valves, if worn, may have to be replaced by new ones, while those fitted with packing rings must have the rings spread. Poppet valves are inspected for tightness on their seats, and the seats are examined, if loose. Since, with double seated valves, it may happen that the distance between them is a little more, or less, than the distance between the faces of the seats, leakage may occur through either the upper or lower valve.

An adjustment is made either in raising or lowering one of the seats, or in decreasing or increasing the length of the distance piece between the valves.

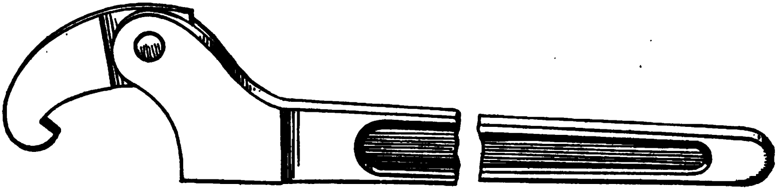


FIG. 2,283.—Adjustable spanner wrench. This type of wrench is especially useful around pumps or engines having different sizes of screw cap stuffing boxes. The movable jaw is adjustable from one size to another with one hand and without releasing the grip on the handle.

The attention given to cross head and connecting rod is mainly directed to taking up the slack in the bearing brasses.

In the latter part of the voyage the engine will have given indication of too much slack by severe pounding and knocking, and the engineers will generally know how much fitting is required.

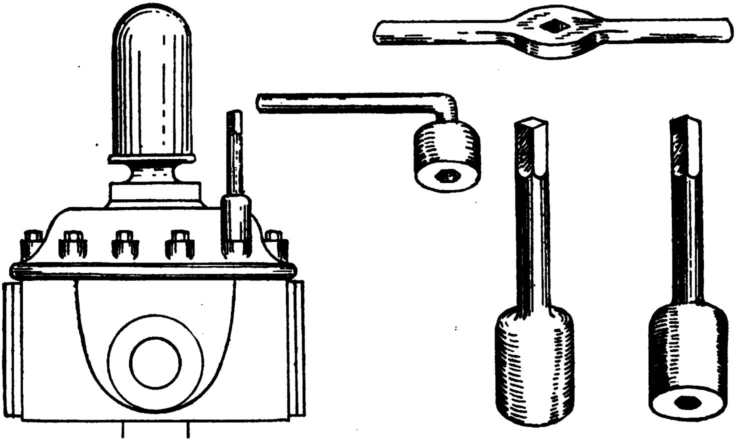
If much wear has taken place, re-babbiting may be required, where anti-friction metal is used.

In the readjustment of the slipper guide care is taken to see that the faces of the guide remain exactly parallel to the center line through the cylinder axis and the center line of the crank shaft. In larger repairs, where a cut has been taken off the face of the guide in the planer or milling machine, an accurate lining up is employed to secure this parallel position.

To meet the wear, either the guide may be brought toward the center by shims or distance pieces, or the slipper may receive the shims, or have higher Babbitt slips or blocks fitted in the dovetailed recesses. The backing strips are then adjusted to the new position, or thickness, of the slipper. The cross head guide in the horizontal river engines may require a more extensive overhauling, as the boat, after severe service, may change its shape considerably.

The distortion of the main axis, through the center of the cylinder and the center of the crankshaft, may be so great that a relining of the cylinder, as well as of the cross head guide, may be necessary.

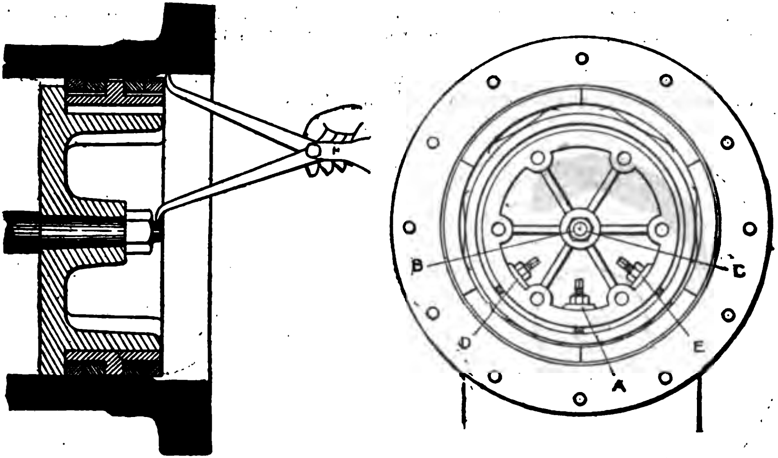
In beam engines the side pressure on the guide is generally so small that an adjustment of the slippers is seldom required. The result is then often due more to deformations of the gallows frame, and to correspondingly changed position of the walking beam.



FIGS. 2,284 to 2,288.—The socket wrench and its use. There are places where a Moncky wrench cannot be used to advantage, for instance, in removing the nuts from the discharge chambers of many pumps, as here shown. A socket wrench is the thing to use although an S, or straight handle wrench could be employed. Unless the jaws of a Moncky wrench have a full bearing on the nut they will spring and this eventually ruins both the nut and the wrench.

The bearings in the cross head and in the ends of the connecting rod must be adjusted, not only to fit on their pins, but also so that they may preserve the right position of the piston in relation to its bore.

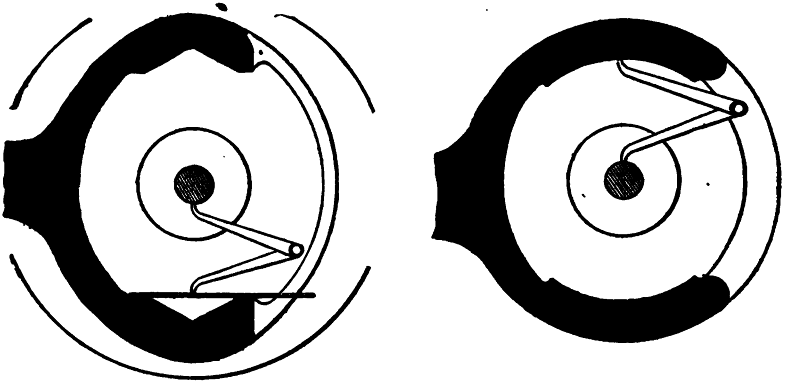
The piston must preserve the right clearances at top and bottom, and, upon wear in the bearing, a lining up by distance pieces must be resorted to, so that the piston will return to its original position. To realize the desired amount of clearance quickly by the adjustment, some engineers employ trammels to show how far a certain mark on the piston rod or cross



Figs. 2,289 and 2,290.—***Centering a piston.*** The tools necessary for centering the piston are, a scale, preferably a 12 inch scale, and a pair of inside calipers, together with the necessary wrenches for removing the cylinder head and follower plate and for turning the adjusting screws in the spider. After removing the cylinder head, have the crank turned to an approximate dead center nearest the cylinder. This will bring the piston to the end of the counterbore in the cylinder as in fig. 2,289 where it may be easily reached. After removing the follower plate, the interior of the spider will be brought into plain view. In many instances the end of the rod will be found to project a trifle beyond the face of the nut. In this case open the calipers, and, placing one leg against the projecting end of the rod and the other against the counterbore, take the distance between the rod and the counterbore, being careful to have the extremities of the legs parallel to the face of the spider as shown. The reason the counterbore is to be preferred to the cylinder wall is that it is not subjected to wear and hence it retains its true cylindrical form. Place the calipers, first at the bottom, then at the top, ascertaining the distance by means of the scale in both instances. Then set out or compress the legs of the calipers an amount equal to one-half the difference between these two measurements, that is, if the bottom measurement is found to be, say, 6⅞ inches and the top measurement is 7 inches, then the calipers would be set to measure either 1/16 less than 7 inches or 1/16 more than 6⅞ inches, because the difference is seen to be ⅛, one-half of which is 1/16 of an inch. After setting the calipers loosen the jamb nut on the center adjusting screw A, fig. 2,290, at the bottom of the spider if three screws be used, and on both of the screws if only two are found. Turn the center screw to the right a very little and then place the calipers on the rod, as in fig. 2,289, to see whether the spider has been raised enough. If not, give the screw another turn and again try the calipers, continuing in this way until the spider has been raised to the proper position. Now place the calipers at the sides of the rod (B and C), to see whether the piston is centered sidewise. If it be not exactly in the center, turn one of the side screws D or E, until it is centered. Turning the side screw will also have a tendency to raise the piston slightly, so the calipers must again be placed on top of the rod to see that it is not too high. If it be too high, the center screw must be turned back a very little so as to lower the piston. The calipers are to be placed, first at the top and bottom then at the sides, and the screws in the spider are to be turned very carefully until the end of the rod occupies a perfectly central position. Before putting on the follower plate, caliper the rod very carefully all around the cylinder (counterbore). If the rod be exactly in the center of the cylinder the calipers should just "feel" the rod at all positions around the counter bore. When tightening the jamb nuts on the three adjusting screws care should be taken to see that the screws do not turn either way.

head should stand with relation to a convenient, unchangeable face on the cylinder casting. Others prefer to put marks on the guide and on the slippers, which coincide at the extreme positions of the piston.

The wear in the brasses is taken up by reducing the thickness of the distance pieces between the two halves of the bearing.



FIGS. 2,291 and 2,292.—*Centering the piston rod.* If the engine be fitted with guide bars, or integral planed guides, have the cross head moved to the end of the guides nearest the crank. Then place the scale across the guides at the end nearest the cylinder and at a point where no wear has taken place. With the inside calipers take the distance between the scale and the underside of the rod close to the gland so as to get as great a distance as possible between the points at which the measurements are taken, for where wear occurs, the diameter of the rod will be found to vary, the effects of which must be obviated as far as possible. Care must be taken not to spring the scale, and also to have the ends of the legs of the calipers at right angles to both the rod and the guides. The calipers should be adjusted so as to just "feel" the rod when moved back and forth on the scale. This position of the calipers is illustrated in the figures. The scale is then to be moved to the opposite end of the guides which will be the end nearest the crank. Place the scale across the guides at a point about two inches from the farthest point on the rod reached by the gland. Then lower the upper shoe of the cross head, if it need lowering, and raise the cross head until the calipers will just pass under the rod, but not as easily as at the opposite end of the guides. Again bring the scale to the cylinder end of the guides and try the calipers. It is very probable they will not touch the rod at all. If not, set them so as to just "feel" the rod and then try them again at the crank end, having the cross head raised until the calipers just touch the rod. Set up the upper shoe until it has an easy fit against the upper guide. This may be determined by placing a piece of drawing paper on the upper shoe. Then set it up just tight enough to allow the paper to be withdrawn without tearing it. It is presumed these adjustments, both of the piston and of the cross head, will be made while the engine is cold, or practically so, so that when the frame is again warmed by the steam this amount of lost motion will be slightly increased. It is not a good plan to attempt to center a cross head by taking measurements both above and below the rod, thinking that the rod should lie exactly midway between the two guides. While this may be found to be true in one engine, there will be found many others in which it will not work. When centering the piston rod of an engine having bored guides, it will be better to caliper between the center (crosswise) of the upper guide and the rod, as in fig. 2,292, if the engine run over, and between the center of the bottom guide and the rod when running under. The object of this is to avoid using surfaces, especially in old engines, that are or have been subjected to much wear.

Frequently, thin shims of sheet metal are taken out, until the bearing clamps the pin when screwed or keyed down hard. The shims are then replaced in sufficient thickness to realize the desired clearances. Thin shims should be well secured to prevent them getting adrift and falling out under motion of the engine.

The adjustment of the brasses should be such that the connecting rod will continue to swing in a plane through the axis of the cylinder and square to the faces of the slipper guide. With divided upper connecting rod brasses, extra care is necessary to realize this result, as well as to make sure that both bearings, or both links, receive their full share of the bearing pressure.

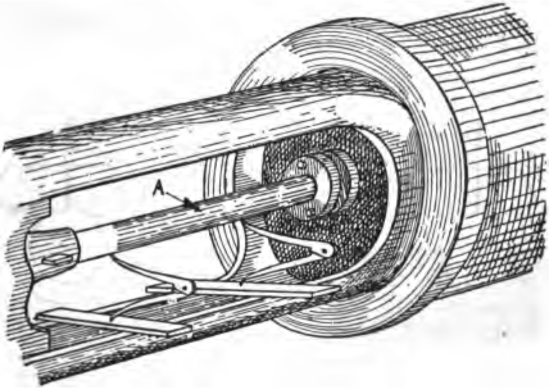
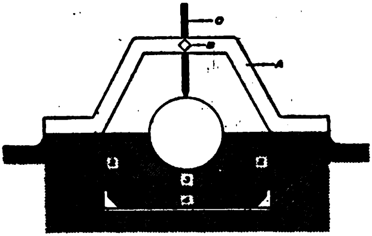


Fig. 2,293.—Alignment test of piston rod. The first step toward success and economy in the use of rod packing is to see that the rod is smooth and that it is exactly in line. The vertical alignment of the rod can readily be ascertained by means of a straight edge and a pair of inside calipers. Place the crank on the dead center farthest from the cylinder. Lay the straight edge flatwise across the guides, and with the calipers obtain the distance between the rod and the straight edge close to the stuffing box. Then find where the rod stops when entering the stuffing box, and at a point corresponding to the depth of the stuffing box, as at A, try the calipers. The cross head should be raised or lowered until the calipers just touch the rod at both these points. It is not desirable to caliper at the extreme end of the rod where it enters the stuffing box, because if grit has worked into the packing for months, perhaps years, the rod will often be found smaller at the point where it reverses its motion in the packing. And the entire length of the wearing portion of the rod may be smaller than at a point next to the cross head, so that the calipers should not be used at that point. The points mentioned are those where the wear is usually uniform and the rod of the same size.

In beam engines the walking beam should swing truly in this plane, and adjustment of the pin bearings should continue until this result is accomplished.

The shafting must always remain in a straight line, in order to prevent such strains by bending as, under the constant reversals in each revolution, may soon lead to fractures. A careful examination and realignment, in case of wear, will, therefore, be necessary. The axis of the shafting should be square to the line through the center of the cylinder, and also square to the plane in which swings the connecting rod. The crank pins should be exactly parallel to the axis of the shafting.

On marine engines the wear generally takes place in vertical direction, and the task of lining up consists generally in raising the shaft uniformly, or on one end more than on the other. A line of sight, parallel and above the normal axis of the shaft, is usually established in more or less permanent form, and is located by center punch marks on bulkheads or parts of the ship. From these center punch marks and additional sighted horizontal battens the position of the shaft is located by fixed trammels, which should barely touch the top of the shaft. The sight line may be easily corrected for deformation of the ship, and, if once established, indicates the difference between the actual and desired position of the shaft.



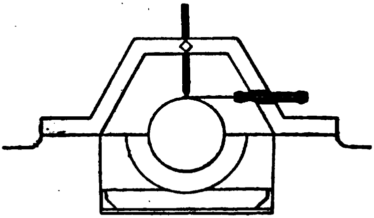


Fig. 2,294.—Tram and method of measuring shaft height.

Fig. 2,295.—Method of using feeler to determine shaft level. The bearing caps are all removed and the tram is placed on bearing No. 1 as shown and the pin set so that it does not touch the shaft. It is then removed to the next bearing and that tested in the same way to determine what adjustments are necessary to bring the shaft to the level with the planed parts of the bed at each bearing. Try all bearings and set tram to the highest point, then go back to No. 1, and by the thickness gauge, or feeler, find the amount that the shaft must be raised.

Each bearing, in its lower half, must be lined up, until the shaft is at its right height. The top brasses are often fitted to the shaft by first measuring the clearance over the pin, when screwed down hard. This is done by little balls or rolls of stiff putty, or by pieces of lead wire, which are squeezed flat by the top brass, so as to indicate, at each end of the bearing, the exact clearance. The reduction of this clearance is then easily accomplished by filing the distance pieces or removing shims.

The accuracy of the lining up of shaft and connecting rod can be tested, also, by successively turning the crank into several positions of a revolution, and by comparing the clearances on each side between crank webs and connecting rod brass. The connecting rod brass should be just free from the pin and free to swing. Several faults may be found:

1. The upper connecting rod bearing, if divided, may have been raised more on one brass than on the other, drawing the lower bearing of the rod to one side, and letting it bear only on the edges, at the same time pressing always against one crank web.

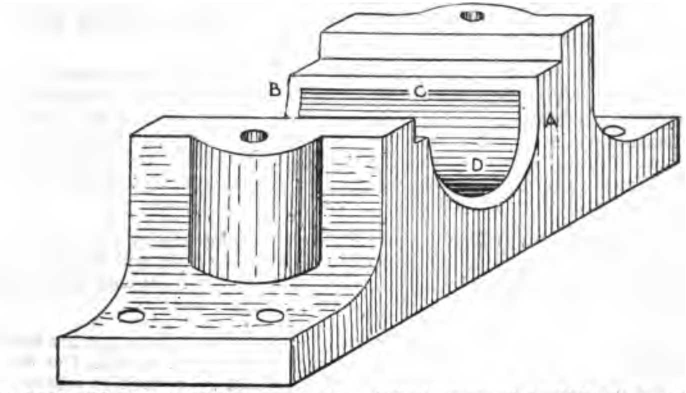


Fig. 2,296.—Method of babbitting a bearing. Babbitt or so called anti-friction metal is composed of tin, antimony and copper mixed in various proportions, and may be purchased, or, if it be desired, it may be easily made. A good mixture, suitable for general use when the duty imposed is light, is composed of fifty parts tin, five parts antimony and one part copper. A harder composition, sometimes termed white metal, is composed of 96 parts tin, 4 parts copper and 8 parts antimony. This mixture is especially suited for journal boxes or bearings, and is mixed as follows: First melt 12 parts of copper and then add 36 parts of tin; 24 parts of antimony are put in and then 36 parts of tin, the temperature being lowered as soon as the copper is melted, in order not to oxidize the tin and antimony; the surface of the bath being protected from contact with the air. The alloy thus made is subsequently remelted in the proportion of 50 parts of alloy to 100 parts of tin. For brass bearings or boxes, a mixture of 64 parts copper, 8 parts tin and 1 part zinc will be found to answer very well; but for bearings not requiring so hard a metal, the quantity of zinc is increased and that of the tin diminished. Bearings that are to be babbitted are usually cast with a receptacle for the babbitt metal, as shown, there being a rib at A, B, and C, forming the cavity D, into which the melted metal is poured. These ribs, in new boxes, are sometimes bored out, or for rougher work may be chipped and filed out to fit the shaft and hold it in line. To prevent the ribs A, B, and C, bearing and cutting the shaft, a piece of pasteboard is laid on ribs A and B, thus confining the journal bearing to the babbitt. The best method is to pour the bearing and then rivet the babbitt well into the cavity D, which is made wide at the bottom to prevent the babbitt coming loose, and then bore out the bearing in the usual manner. As the babbitt metal in a bearing is apt to close across the bore when cooling after being poured, a mandrel of slightly larger diameter than that of the journal should be used to run the bearing on in place of the working journal or shaft. Some mechanics effect the same purpose by wrapping paper

2. Another fault may result when the crank pin swings in a plane, not coinciding with the plane of the connecting rod and cylinder axis. In this case the lower brass of the rod will press against one web in the top position, and against the other web in the bottom position.

When a marine thrust bearing requires attention, make sure that all the thrust collars bear uniformly on the rings, or horse shoes.

With shafts that are out of alignment, it may happen that the collars bear more on the top than on the bottom of the horse shoes, thus tending to tip them, and eventually to break them loose, from the clamping nuts on the side screws. In such a case the shaft must be aligned, or otherwise the whole thrust bearing must be raised on one end, until the face of the horse shoe is again square to the axis of the shaft, coinciding with the faces of the shaft collars.

In the same manner as the piston, the valves also must be retained in an exact position relative to the ports. This is true, since the whole steam distribution depends upon this position. The wear and slack in the valve gear bearing surfaces must, therefore, be taken up with due regard to this position. Trammels are frequently employed, in order to establish and retain a fixed distance between a punch mark on the valve stem and the face of the valve stem guide.

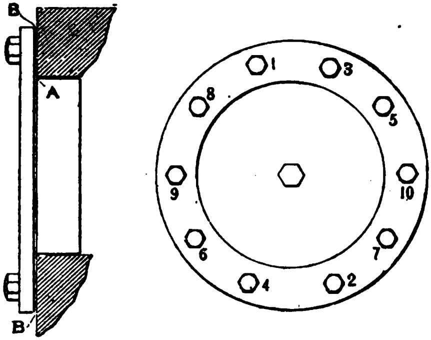
The wear in the link block brasses is generally taken up by placing shims under the brasses. The valve stem end brass, eccentric rod brasses and eccentric straps are generally fitted with shims, to allow an easy and rapid taking up.

In beam engines the rock shaft bearing may require taking up; the valve stem arms are ascertained to be secure on the lifting rods, also the dash pots, gear and drop arrangement of Sickles' cut off gear must be inspected for indications of wear, such as would interfere with its correct working.

FIG. 2,296.—*Text Continued.*

around the journal, but it is wrong to use the journal for the following reasons: To get a good, sound, well fitting Babbitt bearing, the metal should be poured as cool as possible, for if it be heated red hot, it contracts so much in cooling that it does not fit well in the box or frame of the machine. On the other hand, unless the metal is well heated, it is liable to cool and set too soon and thus become unsound. In remedying this the journal or whatever represents it, must be heated and the heating is very apt to bend it. It is obvious then that instead of the journal or shaft, a temporary iron bar of slightly larger diameter than that of the shaft or journal, should be used. The bearing into which the metal is poured, as well as the mandrel or bar of iron should be heated to about 200° to 300° Fahr., which will enable the Babbitt metal to be poured less hot than would otherwise be permissible, thus reducing the contraction of the Babbitt in the bearing to a minimum. A little powdered rosin sprinkled in the box will help the Babbitt to flow easily, and smoking or chalking the mandrel is also said to make the metal run freer and gives a smoother face. As soon as the Babbitt has well set the temporary journal or mandrel should be revolved to free it. To prevent the metal running out of the bearing, its ends are closed by means of either clay, putty or red lead closely packed against the bearing ends and the mandrel. In pouring the melted metal it is best to pour it on the top of the mandrel and let it run down its sides into the cavity of the bearing; brass mandrels are said to be better than iron to pour metal around. In pouring the metal on top of the mandrel it heats the mandrel equally and prevents it bending from unequal expansion, as it would do if it met the heated metal on its lower half only. It may be easily seen that if the mandrel bends, the bore of the bearing will not be cast in line; hence the shaft will bear at the end only and will require to wear the Babbitt down to a bearing.

Valve setting is resorted to when the valve does not perfectly perform its steam distributing function. Indicator diagrams, taken previous to disassembly, will give a possible clue to the desirable adjustments.



FIGS. 2,297 and 2,298.—Tightening a cylinder head. All joints should be pulled up square and even all around from start to finish, especially where a metal joint is used. Dirt being left on joint surfaces often causes leaks, because the two cannot be brought evenly together, and just as often the leak is caused by the uneven strain on the bolts. Take, for example, the cylinder head shown in the figures, which has a shoulder all around the inside of the flange. It will be seen that by pulling on one nut first, the head could be tipped out of true, and only one edge of the shoulder joint would touch. When first starting to set up on the nuts, a good method to follow is to set up on No. 1 nut lightly until the surfaces of the joint meet, then take up the same on No. 2 nut opposite to No. 1, then Nos. 3 and 4 in succession, after which the nuts can be taken up the same amount in the order given. Then go over them all again in the same order until the joint is tight. The space B, will be equal all around if the pulling up has been properly done. This rule applies equally well on all joints, taking any nut for No. 1 and making No. 2 come opposite.

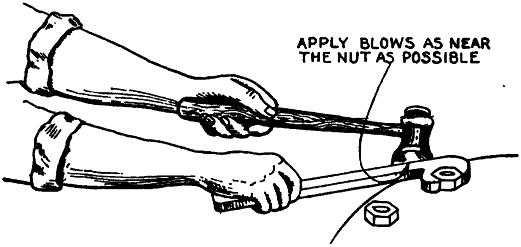
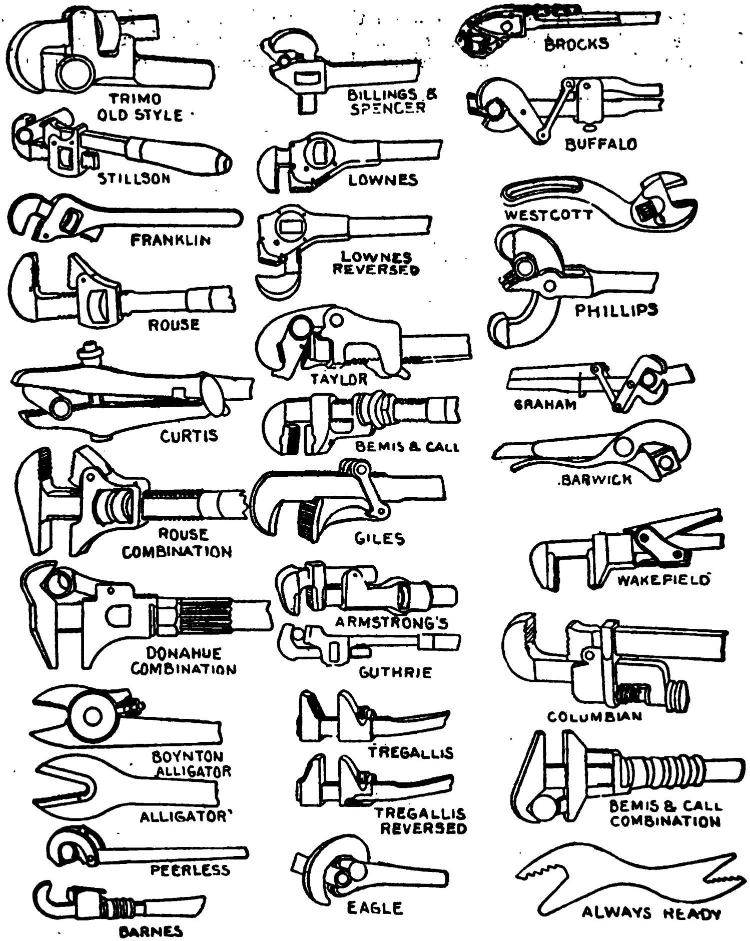

APPLY BLOWS AS NEAR THE NUT AS POSSIBLE

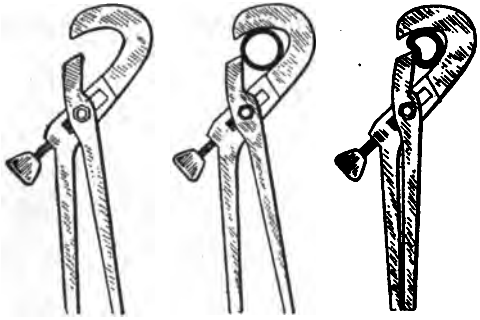
FIG. 2,299.—Starting an obstinate nut or bolt. Rusty, or large nuts or bolt heads often require more than a straight pull. A sharp blow with a hammer often starts an obstinate hold, where a straight pull would not. It is not advisable only in extreme cases to use the hammer on the wrench, but a hardwood block will do as well. In extreme cases a steady pull aided with blows will do the work. *The blow should be delivered **as near the nut as possible**,* as shown in the figure, *instead of at the other end of the wrench as is usually and erroneously done,* thus avoiding the spring and inertia of the wrench, and delivering the full energy direct to the nut.



FIGS. 2,300 to 2,331.—Various pipe and combination wrenches. There are besides these, a few styles that are fairly well known, and several hundred, the majority of which have fortunately never gotten beyond making the original model and patent. *Criticising* these wrenches, a well known dealer says that in his judgment, the world would be just as well (and the inventors better) off, if at least fifteen of them had never been discovered. Judging from the illustrations, the author believes a much larger number could be eliminated without causing any inconvenience to the users of wrenches.

The steam cylinders of the independent auxiliaries should be examined for tightness of their pistons in the barrel, with setting out of old or fitting of new rings. All the working parts are well overhauled, taking up the slack in the different bearings of the connecting rod, crank shaft and valve gear, and all parts are well secured against the possibility of getting adrift during the voyage.

The valve gear of the reciprocating auxiliaries are inspected to determine whether the pumps keep their right full stroke, or whether their tightness is sufficient to hold the steam well. The valve gears of the revolving auxiliaries are examined for the position of the valve on the rod, the position of the eccentric and the tightness of the valve.



FIGS. 2,332 to 2,334.—Pipe tongs and a common result from using them. It is somewhat unusual at the present time to find a kit of pipe fitters' tools containing pipe tongs. Pipe wrenches are so much more serviceable and convenient that tongs have become obsolete. Pipe wrenches embodying the principles of the Moncky wrench and the Stillson wrench are preferable for general work. These are to be had of all dealers in mill and steam plant supplies, and the variety of constructions is large enough to suit everyone. For pipe and tube work an adjustable wrench will usually be found preferable, since a slight motion of one of the jaws will enable the wrench to take hold where fixed jaws might prove useless. The same difficulties are experienced, and often noticed in using pipe wrenches as with those for turning nuts, viz., one wrench is made to serve too many uses. When properly used, a wrench of any size is not adapted to a very wide range of pipe.

NOTE.—Pipefitters often handle Stillson wrenches in a manner destined to ruin the pipe. In screwing up or slacking off on a pipe, always catch the wrench as close up to the thread as possible. Many cases of split pipe have been attributed to the wrench being held at the middle of its length, allowing the pipe to twist under the heavy strain and split the seam. Another source of trouble with Stillson wrenches originates from constantly taking hold of the pipe in the same place. When many hard pulls are necessary to set up, the result is a pipe cut through in places. The proper thing to do in taking holds is to move the wrench along the length a little and back again, so that the teeth of the wrench will not grip twice in the same place. A Stillson wrench should also be set down on its work, so that the jaws will take hold with the work well up in them. There is one thing which limits this, however, and that is the amount of pull which the hold must stand. The stronger the pull, the farther up on the work the jaws must be in order for the teeth to take hold. For this reason also, when making a hard pull, it is not advisable to use a very large wrench on small pipe, as the larger teeth may cut through the pipe or crush it. A Stillson wrench is not so liable to crush pipe as a pipe tongs, and for this reason the former is the best to use. It is best to use a chain wrench on the larger sizes of pipe, discarding the Stillson for sizes over 3 inches. Never use a Stillson on a bolt head, nut, stud or finished work, as there is always a way in which these may be handled with standard or special wrenches.

4. LAYING-UP

The term *laying up*, as applied to machinery, means putting it in such condition that it may remain unused for a more or less extended period of time without impairing its further usefulness.

The principal danger, threatening to impair the future usefulness of the machinery, consists in rusting and corroding. The engine must be thoroughly protected against rust and corrosion.

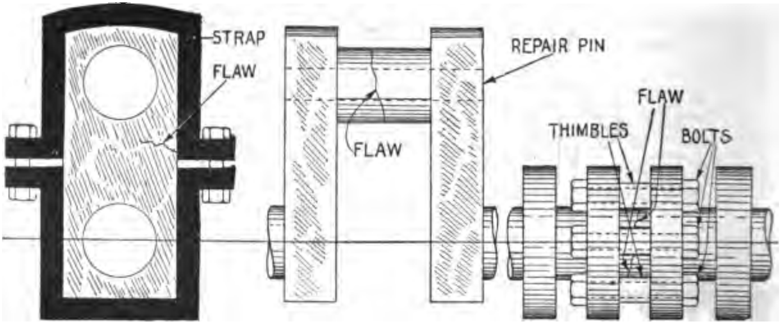


FIG. 2,335.—Repair of flaw in ***crank web.*** Two iron or steel strips, say about 2 inches thick, are heated and shrunk on, then bolted as shown. The bolts should be as large as possible, for ordinary size of shafting about 2½ inches diameter, but larger than this if they can be obtained. Coupling bolts would do very well in most cases.

FIG. 2,336.—Repair of broken ***crank pin.*** Bore out the crank pin to almost one-third its diameter, and put in a repair pin, a driving fit. A small locking pin screwed half into crank pin and half into repair pin will keep it in place.

FIG. 2,337.—Repair for broken ***thrust shaft.*** The collars should be bored or slotted out and bolts with thimbles fitted between the collars where the flaw or break is situated. The thrust block rings next the broken part will have to be removed.

Particularly on the finished surfaces, such as piston and valve rods, cylinder bores and valve faces, as well as on all the bearings, a great impairing of the future usefulness of the engine would take place, if rust should deprive them of their smoothness and condition of least friction.

The remedies applied are, in this case also, protecting coats of paint, oil or tallow, and the prevention of condensation by ensuring a dry surrounding atmosphere.

All rough surfaces may be painted, while the finished surfaces receive coats of heavy oil, tallow, or a mixture of tallow and white lead, which latter adheres very firmly and possesses at the same time considerable body and lasting qualities.

The copper, brass or bronze parts are naturally more immune from corrosion than the steel parts, and the latter, therefore, require the greatest attention and an occasional inspection, to see that the protecting coats have not been rubbed off.

The bearings of the shafting may hold moisture that would corrode the lower part of the shaft, if plenty of tallow be not applied to the lower brass.

An occasional turning over of the engine with the hand turning gear is a wise precaution to make sure that nothing rusts tight.

The greatest care should be taken to remove all water and moisture from the engine, and no protective coating should be applied to any part of the machinery that is not dry and at least moderately warm.

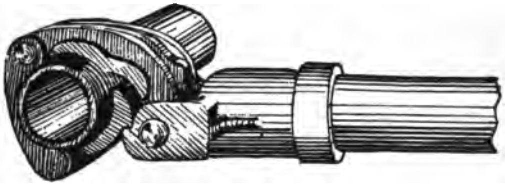
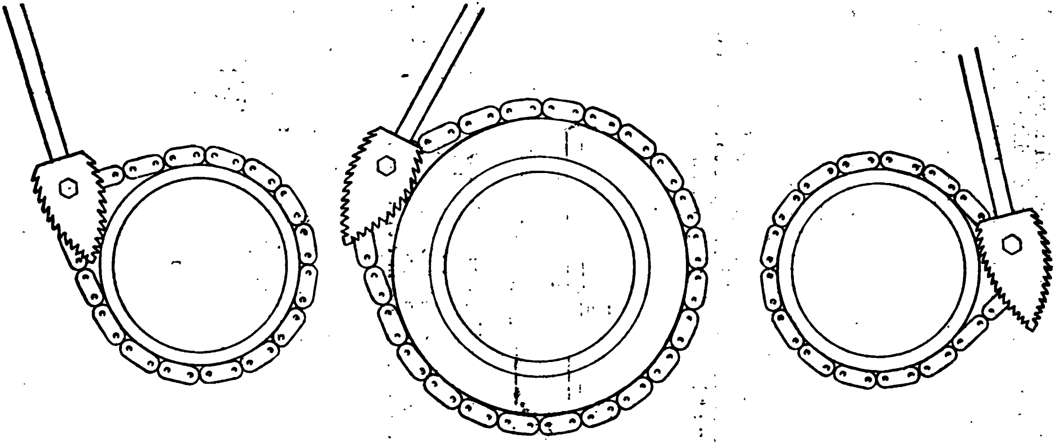


Fig. 2,338.—Strap wrench for polished pipe, thin tubing and rods. The type of wrench should be employed on such parts as piston and valve rods, eccentric rods, brass and nickel plated piping, polished railings and tubing. While this type of wrench is applicable to all pipe surfaces, it is particularly adapted to very smooth and all polished surfaces of rods and pipes. Even when emery cloth or thin leather is used in turning polished piping, the ordinary pipe wrench with its sharp teeth is liable to cut through and mar the surface, especially under a heavy pull.

All drain cocks should be kept open, and the engine turned several times for the complete removal of all water. If a systematic inspection of the ports or any pockets of cylinders, liners, valve chests, stop and other valves should reveal any water lodged in these pockets, it should be sponged out or a little hole drilled for drainage. In localities where freezing weather is likely to be endured by the laid-up machinery, this proves of the greatest importance, as such imprisoned water, upon freezing, expands, and is very apt to crack the casting or the part in which it is contained.

The auxiliary machinery and piping are prepared for laying up in a manner similar to that used with the main engine.

All rough parts may receive a coat of paint or linseed oil, while the finished parts receive a coat of heavy oil, of tallow, or of a mixture of tallow and white lead.



FIGS. 2,239 to 2,341.—Application of chain wrench. These are especially adapted to large piping flanges. The principal point to be observed in using chain tongs is to get a good bight before attempting to pull, and to place the serrated edges against the pipe in such position as to reduce the tendency to crush the pipe. When removing fittings from pipe it is preferable to secure the fitting by screwing in a piece of pipe and using the tongs or wrench on the pipe, instead of on the fitting to be removed. A better hold can be obtained upon the wrought iron pipe than upon the cast iron fitting.

The drainage of water should be as thorough as for the main engine. It is better to drill small holes or to take down the valves or parts than to allow water to lodge in any pockets of the condenser, pump cylinders, pipes, valve chests, manifolds, etc.

A great deal of expense, in way of repairs, can be avoided by faithful attendance to these precautions.

The preservation of the efficient and useful condition of the machinery depends upon the successful protection against condensation of moisture. There are two factors generally influential in preventing the precipitation of water from the atmosphere: The first is to keep the air as dry as possible by preventing its absorption of moisture from water in the bilges or open tanks.

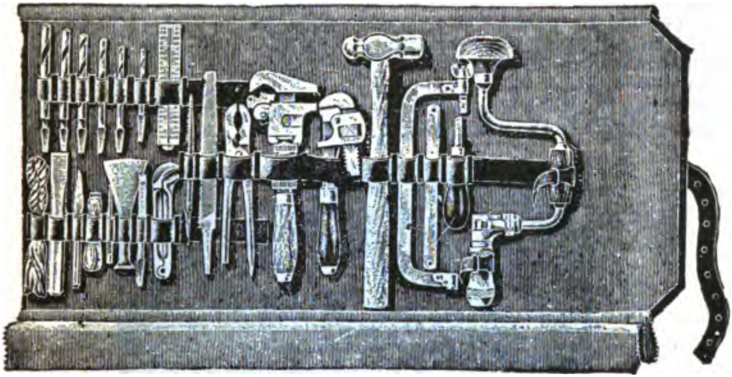


FIG. 2,342.—Kit of tools useful on repair jobs. With this outfit various adjustments and repairs can be made such as those arising from the ordinary mishaps likely to be encountered in operation.

NOTE.—***To keep machine bright.*** Take one ounce of camphor, dissolve it in 1 lb. of melted lard; take off the scum, and mix as much fine black lead as will give it iron color. Clean the machinery and smear it with this mixture. After 24 hours rub clean with soft linen. cloth. It will keep clean for months under ordinary circumstances.

NOTE.—***The rust preventive*** composition of Jones & Co., Sheffield, is a composition of wax, fat, turpentine, and small quantities of iron oxide.

NOTE.—***To separate rusty pieces.*** By boiling the object in petroleum, success is certain. It is necessary to treat them with alcohol or spirit to avoid subsequent oxidation, petroleum being in itself an oxidant.

NOTE.—***A patented process*** to prevent rusting of wrought or cast iron consists in applying with a brush a strong solution of potassium bichromate and drying in a stove or over an open fire. Drying at ordinary temperature is not sufficient. To ascertain if the heat be strong enough the iron is moistened with a little water. So long as this takes up any color the heat must be increased. When the proper degree of heat is reached a fine deep black layer results, which is not acted upon by water, and protects the surface from the action of the atmosphere.

As is well known, air of low temperature can hold suspended moisture less easily than warm air, which tendency finds its climax in freezing weather, when the moisture is nearly all thrown down in form of frost or ice. By keeping fairly dry air, at a relatively high temperature, the dangerous "sweating" can be largely avoided. In localities where the laid up machinery is exposed to freezing temperatures it is found advantageous to provide for artificial heat, in some form, in the engine and boiler room. This helps to keep the temperature of the air and the machinery as nearly uniform as the conditions will allow. On the other hand, too high a temperature, say above ninety degrees Fahrenheit, may be undesirable, as it may stimulate larger evaporation from any water that may be present, and thus add to the total humidity of the air.

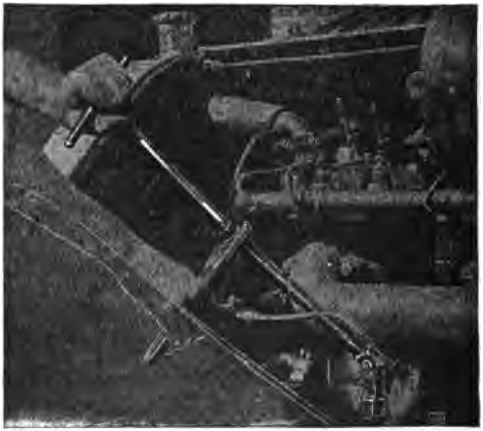


FIG. 2,343.—Application of the socket wrench to an inaccessible nut on an automobile. The development of the socket wrench is due largely to the automobile, and with the great variety handles, rachets, etc., now on the market, the most inaccessible nut is easily reached. The illustration shows a socket coupled to a Mossberg ratchet, handle, and extension bar.

The second is to keep the air, as uniformly as possible, at an average comfortable temperature which will prevent sweating or throwing down of the moisture.

It is apparent that the accumulation of water as upon the floor or in tanks furnishing extended surface areas for evaporation would furnish considerable moisture conducive to the formation of rust. Accordingly, floors and other surfaces should be kept dry and water tanks should be kept empty, or, if water be needed, should be supplied with tightly fitting covers, in order to prevent evaporation.

CHAPTER 43

ROTARY ENGINES

Considerable time, thought, and money have been expended on the problem of devising a machine to compete with the reciprocating engine, which would avoid the disadvantages of applying power alternately in reverse directions. The efforts, in most cases, could have been more profitably spent if directed in other channels.

The so called rotary engine may be defined as *an engine in which the piston revolves with the shaft, or the cylinder revolves upon the piston.*

The distinction between a rotary engine and a turbine is that *the rotary engine employs* **both** *rotating and reciprocating motions, while the turbine operates with a rotary motion* **only.**

The length of the stroke, therefore, is the path of the piston, around the cylinder bore. In some forms of rotary engine, cam shaped pistons, are employed; in others, the piston head carries small sliding pieces, which are moved in and out, to pass an abutment by cam motions; in some cases, the abutment itself is moved, to allow the piston to pass.

Fig. 2,350 illustrates the principle of operation of the ordinary types. Steam enters continually at S, and escapes at E. Sliding pistons P, P′, project from the rotor R, and touch the casing C, furnishing the moving surface upon which the steam force acts. Some simple contrivance, such as a circular groove in the

FIGS. 2,344 to 2,349.—*Some early rotary engines.*

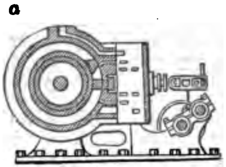


FIG. 2,344.—*No. 1.* It has a single piston keyed to the hub and rotating in an annular chamber, which has the function of a cylinder. In the middle, on the right, is the abutment which slides radially to allow the piston to pass. Above and below the abutment, respectively are the induction and eduction ports.

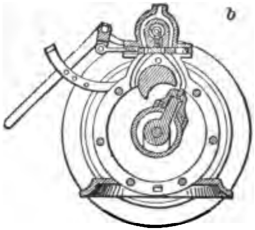


FIG. 2,345.—*No. 2.* It has a single piston which passes a crescent-shaped rocking abutment situated between the induction and eduction ports.

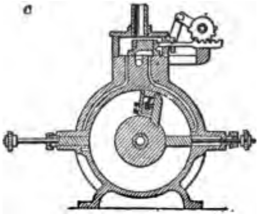


FIG. 2,346.—*No. 3.* The piston revolves on a hub. concentric with the cylinder, and the annular steam space between the hub and the cylinder side is traversed on each side alternately by sliding abutments, connected together and operated by a segmental cam on the piston shaft, which impinges against anti-friction rollers of the frame.

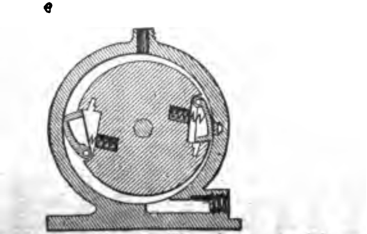


FIG. 2,347.—*No. 4.* The piston wheel is arranged eccentrically within the cylinder, and has two buckets, which are expanded radially by springs, and withdraw to pass the abutment by contact with the cylinder. Packing segments on the piston wheel and the edges of the buckets confine the steam.

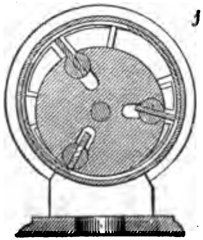


FIG. 2,348.—*No. 5.* It has three pistons, which have a certain freedom of motion in seats in the inner cylinder which rotates in an eccentric drum.

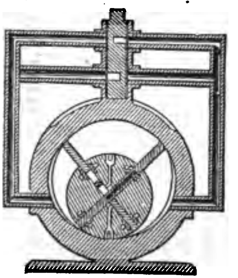


FIG. 2,349.—*No. 6.* It has diametric pistons, which are equal in length to the diameter of the casing, and slip to and fro in slots in the eccentric hub.

ends of the casing and projections on the pistons, must be added to keep the latter in contact with the abutment.

This particular scheme has the fault that it makes no provision for expansion of the steam, which puts it in a class with the small direct acting steam pump.

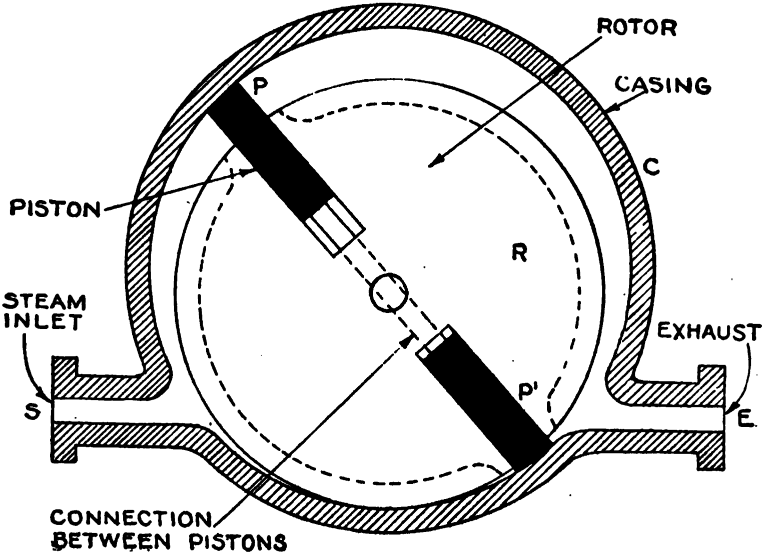
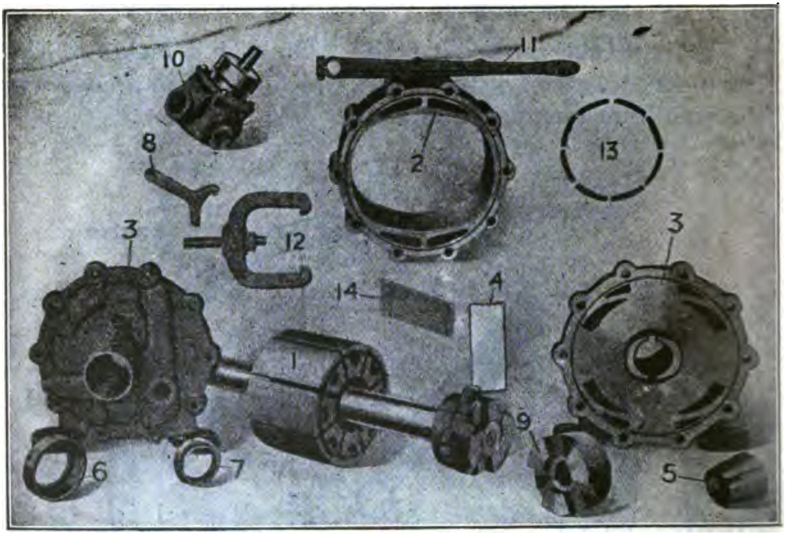


FIG. 2,350.—Double piston non-expansive rotary engine illustrating working principles. ***In operation,*** steam enters at S, and exhausts at E; during its passage, the force of the steam acts on the pistons P, thus causing the engine to rotate. This particular type makes no provision for expansion, hence the waste of steam is excessive.

Inherent difficulties have prevented the economic success of the rotary engine, while the development of the steam turbine has satisfied the demand intended to be met by the former.



FIGS. 2,351 to 2,364.—Parts of Soule non-expansive double "steam feed" or rotary engine: 1, runner; 2, case; 3, heads; 4, pistons (8 in set); 5, taper sleeve bearings; 6, packing nut; 7, follower ring; 8, spanner wrench; 9, coupling; 10, valve; 11, valve handle; 12, pull off clamp and screw; 13, packing rings for heads, or runner (8 in set); 14, packing wires for pistons (16 in set). ***In construction,*** the runner is in effect a plain cylinder, mounted upon a shaft, and has 8 radial slots cut through its periphery down nearly to the shaft, as shown. The pistons consisting of flat pieces work in these slots. The interior of the case is a modification of an ellipse in form, and the "runner" is just an easy fit in the smallest diameter of the "case." The heads have each four ports, changing from steam to exhaust. The shape of the interior of the case, between the ports, is an arc of a circle, and the distance between these ports is ⅛ of the circle, corresponding with the distance apart of the pistons. It will be seen that the pistons are at work (that is, have pressure upon them) only while they are between these ports; hence while thay are subject to pressure they are not moving in their slots, creating friction and consequent resistance and wear. While they are moving out and in they are in equilibrium, either in the steam or exhaust space. There is always one piston between each steam and exhaust port, to be acted upon by pressure, and as it is also ⅛ of the circle between the smaller ends of the ports there is always likewise one piston between the ports there, to prevent the steam passing backward; hence no need that the runner make a tight joint with the case. The construction gives a direct rotary impulse to the runner, the same as two opposite points, eliminating pressure or strain on the journals except the tortional strain due to the power imparted to the shaft. The small pipes answer the double purpose of keeping steam inside of the pistons to throw them out and to convey the oil to the inside of the machine. The oil first passes from the small pipes to the journals, thence it is carried by the steam into the center of the runner, which is hollow, and is distributed to the sides of the pistons, by which it works out to the case and is carried out through the operating valve by the exhaust steam, thus lubricating the entire machine by means of only one lubricator. If the steam feed is to drive a rope drum, connect the feed to it with a gear wheel and pinion so that it will give the feed about one revolution to each 18 inches travel of carriage.

Although possessing the advantages of a non-reciprocating motion, and cheapness of construction, all rotary engines are more or less wasteful in fuel. This results from several causes, not the least of these being the tendency to excessive leakage. Another loss is caused by radiation from the comparatively large portion of metal surface in which the steam comes in contact. Further, since the machine develops its power by the combination of a small driving force with a high velocity, the rubbing friction due to an attempt to keep the piston tight will absorb an excessive amount of power.

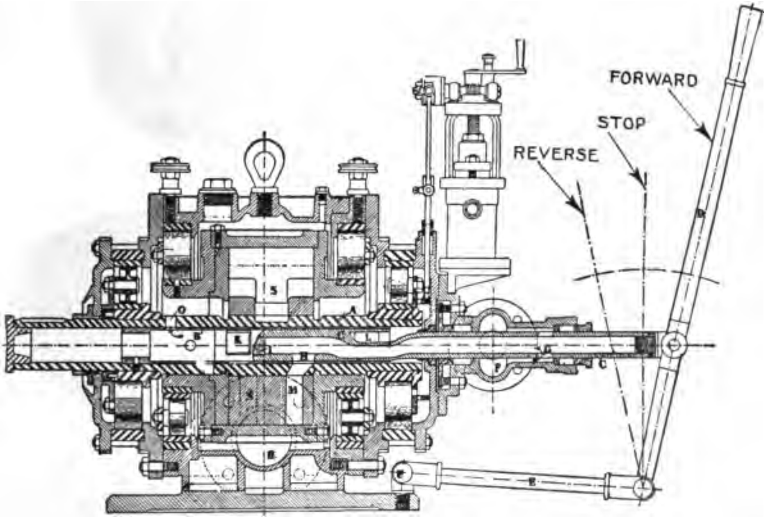
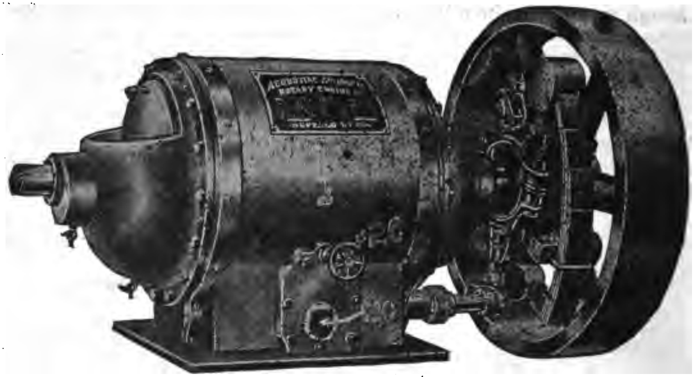
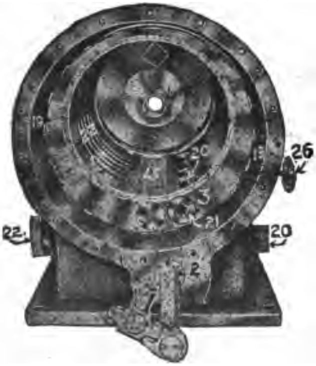


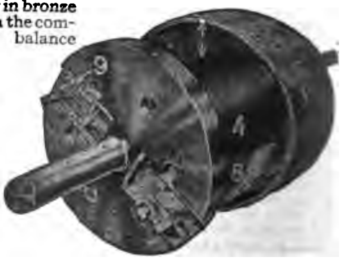
Fig. 2,365.—Hult reversible rotary engine. ***In operation,*** live steam passes from box F, through orifice G, into distributing tube C, pierced by slots H, which by longitudinal movement of the tube can be brought in line with steam passage M or N, corresponding to forward or reverse motion. Tube C being closed on top by stop I, steam can only reach the cylinder through slots H. Exhaust steam escapes through orifice K, or L. As shown, slot H, opens fully into passage M, admitting steam for driving forward; exhaust passes through N, and, when N, reaches K, into the space outside stop I, in distributing tube; it then passes through O, into the space outside the cylinder and finally to exhaust pipe R. When lever D, is set for *reverse*, slots H, come opposite N, for *admission*, M, in this case serving for *exhaust*, each time it opens into L.

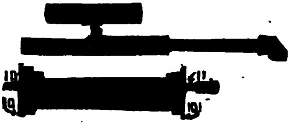
These unavoidably bad features overbalance the good points, except possibly in certain special applications of power where economy of fuel is of secondary importance. The rotary engine is useful where it is desired to develop a large amount of power in a small space, and under conditions requiring satisfactory working with little and incompetent attention.





Figs. 2,366 to 2,373.—Detail views showing construction and operation of the Augustine rotary engine. The rotating portion is made up of a cylindrical piston 4, of fig. 2,369, which carries sliding blades 8, moving in and out on roller bearings 9, in slots in the flanges. The blades are guided by arms 11, of fig. 2,373, which bear in bronze bushings in the compensating balance





The only class of rotary engine worthy of serious consideration from the view point of economy is that in which steam

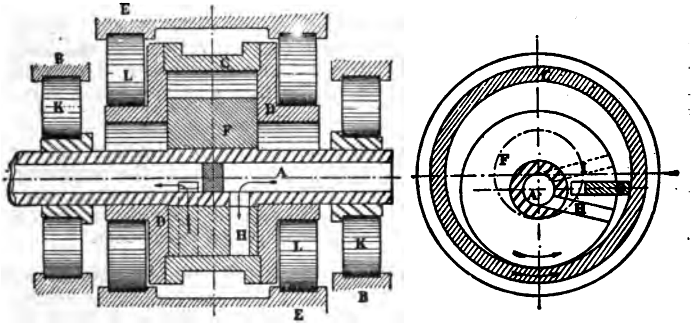


FIG. 2,375.—Hult single door rotary engine. ***In construction,*** the piston F, placed eccentrically within the cylinder C, is shrunk on the main shaft A, and is provided with passages for the admission of steam H, and for the exhaust I; the sliding shutter G, is placed near the admission passage. The steam from the boiler enters through the hollow shaft A, and the passage H, into the space enclosed between the piston, the cylinder, and the shutter, and pushes around the shutter for turning the piston, the shutter having the same length as the piston. When the piston reaches a certain angle the admission of steam is cut off automatically by a valve placed inside or at one end of the hollow shaft. The volume of steam introduced then works by expansion upon the shutter G, until the pressure becomes low enough, when the rotary motion of the piston, eccentric to the cylinder, uncovers the openings for the exhaust steam. There is no valve for admission or exhaust of steam in the cylinder itself. Centrifugal force acting upon the shutter ensures a steam tight fit between the shutter in the piston and the cylinder, both revolving in the same direction. The shaft with the piston runs on the roller bearing BK. The cylinder runs free on the roller bearing EL; it is drawn round by the piston in consequence of the pressure between them at the line of contact. The piston and the cylinder have thus the same circumferential speed, but in consequence of their eccentricity the angular speed of the piston is slightly higher.

FIGS. 2,367 to 2,374.—*Text continued.*

rings 12, shown in fig. 2,371. These flanges are bolted fast to the piston and have bolted to them the telescopic discs 18, which fit over the abutment 3, shown in figs. 2,367 and 2,368. The plates 10 of fig. 2,373, close the openings through the telescopic discs. The entire rotating member is carried on a shaft which runs in the ring bearings 25 of fig. 2,371, these bearings being central with the outer casing. The compensating rings 12, are so arranged as to balance the centrifugal force of the blades, thus keeping the rotor steady, and they also serve by their positions in the grooves 23 of fig. 2,371, to move the blades in and out in the grooves of the flanges, so that they are kept bearing against the cylinder walls at all times, with a packing strip along the edge to make a steam tight joint. The piston, which is mounted in the abutment 3, eccentric to the abutment bore, provides the means for expansive working of the steam. Steam is admitted through the pipe 20, of fig. 2,366, and is controlled by a Corliss type valve 6 of fig. 2,372, operated by a rocker arm and eccentric, the eccentric position being controlled by an inertia shaft governor. The piston makes contact with the abutment on the priming pockets 13, being held there by the bearings, 25. As the blade passes the admission ports 20 of fig. 2,367, the admission valve is opened wide, and steam is admitted behind the blade, forcing the piston forward until cut off occurs, that point being controlled by the governor according to the load. Steam then expands behind the blade, forcing the piston forward until it reaches the exhaust port, 22 of fig. 2,367, when is worked expansively. An example of this class is shown in figs. 2,366 to 2,373.

In fig. 2,366 is shown the assembled engine ready to run. It has a fly wheel governor which changes the point of cut off to suit the load. The eccentric, by means of an eccentric rod and rocker arm, drives the oscillating admission valve as shown in the figure at the lower part of the engine.

In fig. 2,367, the ports through which the valve admits steam are in the bottom of the casing, one of them being seen where the revolving piston on the rear side has been removed. The ports in the abutment are in line, but the ports in the valve are staggered so that the valve rocks back and forth steam is admitted, first to one piston, then to another.

Two revolving pistons are mounted centrally on the outside casing, as shown in fig. 2,371, and held in this position by bearings in the heads. Each piston has a counter balance blade as seen in the figure, at the lower right hand side of the piston and the counter balance in the slotted arm of the piston at the upper left hand. This blade laps across the flanges of the piston, the object being to make a steam tight joint.

In operation, as the piston blade crosses the lower center of the engine, the valve rocks to open the port and admit steam behind the blade piston; this forces the revolving piston around, and the valve rocks to cut off steam at the point required by the load, as indicated by the governor action. From this point of cut off, expansion takes place, and when the blade on one side reaches its uppermost position, the blade on the other side crosses the lower center, and that piston begins to take steam. Exhaust takes place without the action of a valve when the blade crosses the opening to the exhaust port. As constructed, admission and expansion take up about 320 degrees of a revolution, and exhaust the remaining 40 degrees.

FIGS. 2,366 to 2,373.—***Text continued.***

release occurs. Pressure is equalized between the cylinder and the inside of the telescopic discs through the small ports 17, of fig. 2,368 and small check valves. The pockets 16, are thus filled at all times with the lubrication and condensation of the same pressure as that in the cylinder, and this bearing against the inside of the telescopic discs gives a floating bearing. Oil is also forced out through these ports 17, by centrifugal force and lubricates the running joint between the telescopic discs and the ribs 14. The ends of the telescopic discs make a tongue and groove joint with the center ribs 15, of fig. 2,368, of the abutment, thus preventing leakage. The flanges bear against the ends of abutment 3, making a running joint by means of packing rings. The piston, therefore, rotates inside the abutment, the flanges rotate against the ends of the abutment, and the telescopic discs rotate in the space 19, of fig. 2,367. The endwise pressure is taken by the flanges, which bear against the telescopic discs, so that no pressure comes against the heads 5, of fig. 2,370. As the steam is swept from the admission port around to the exhaust port without reversal, the result is a "uniflow" effect, and the exhaust steam is never carried backward over the cylinder surfaces. The engine can run at high or low speed, with high or low steam pressure, and as there is no compression, water entering the cylinder does no damage.

CHAPTER 44

STEAM TURBINES

The development of the steam turbine was caused chiefly by the demand for a higher rotative speed, due to the introduction of the dynamo. Originally the dynamo ran much faster than the engine, the speed reduction being accomplished by means of a belt drive with pulleys of unequal diameters.

The problem which naturally presented itself to the designers of dynamos and engines was how to reduce the speed of the dynamo, and increase that of the engine so that the two could be directly coupled. This resulted in the introduction of the short stroke or "high speed" engine, capable of running with great rotative speed. On account of the large clearance space, due to the short stroke, these engines were not very economical in the use of steam; this, together with the disadvantages of running reciprocating engines at high rotative speeds caused inventors to turn their attention to the turbine. The problem now was reversed, that is, it became a question of how to increase the speed of the dynamo, and reduce that of the turbine sufficiently for a direct connected unit.

Principle of the Turbine.—The operation of a turbine is *due to centrifugal force* produced by *changing the direction of a jet of steam escaping from a nozzle at high velocity.* This is done by so placing the nozzle that the jet will impinge on numerous curved vanes attached to a wheel free to revolve, thus causing rotation.

The kinetic energy of the steam is considerable, for although its weight at ordinary pressures is very small in proportion to its volume, the velocity of steam escaping from a nozzle is very great. For instance, at 180 pounds gauge pressure, steam flowing directly into the atmosphere has a velocity of about 3,000 feet per second, or 3,700 feet per second when flowing into a vacuum.

The principle of turbine operation may be more clearly presented by aid of a diagram, fig. 2,375. V, V′, are curved vanes attached to the circumference of a disc, or *rotor* W. At one side, and placed at an acute angle with the rotor, is a steam nozzle N. In operation, steam issuing from the nozzle at high velocity has its direction changed by the curved vanes, it being obvious that

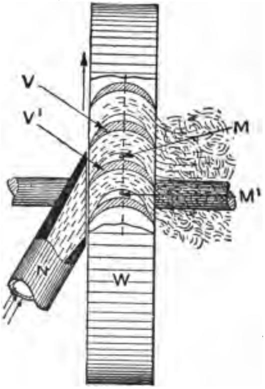


FIG. 2,376.—Diagram showing principle of turbine operation. Steam issues from the nozzle N at high velocity, and on striking the curved vanes its direction is changed; the centrifugal force thus set up tends to cause the wheel to rotate in the direction of the arrow. This action is caused by *impulse* up to the dotted line, and by *reaction* beyond that point to the end of the vane. The usual application of the words "impulse" and "reaction" is erroneous and misleading. The pressure between the vanes is the same as the pressure within the casing in which the wheel runs. With the vanes constructed as shown, there would be spaces M, M′, not filled by the steam, since the area of the passages at these points is greater than at the entrance and exit. Some manufacturers, however, make the blades thicker at the center than near the edges, to maintain a constant area and so avoid possible eddy currents.

if it enter in the direction shown by the arrows, it will leave its course as indicated.

The steam, in flowing through the curved passages, causes the wheel to rotate, due to *impulse* in flowing from the entrance to the dotted line, and to *reaction* in flowing from the point to the end of the passage.

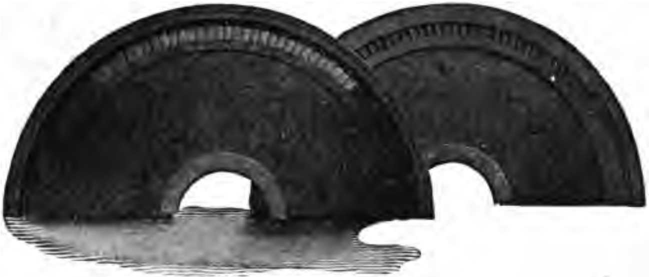


FIG. 2,377.—Front and back of Ridgway diaphragm with cast nozzles. The nozzle construction varies in the various diaphragms, depending on the area required. Nozzles with small area are solid castings of special alloy, machined and accurately shaped. Such nozzles are used in the first stages and are bolted in place. In later stages, the blades forming the nozzles are cast in place in the diaphragm and extend all or part way around the periphery of the diaphragm.

It must be obvious that the rotation of the wheel is caused by centrifugal force acting against the forward vanes, and *not by the pressure of the steam.* For, if the curved passages be closed, thus bringing the steam to rest, it would press equally against the vanes, V, and V′; since these present equal areas, there would be no excess force tending to cause rotation.

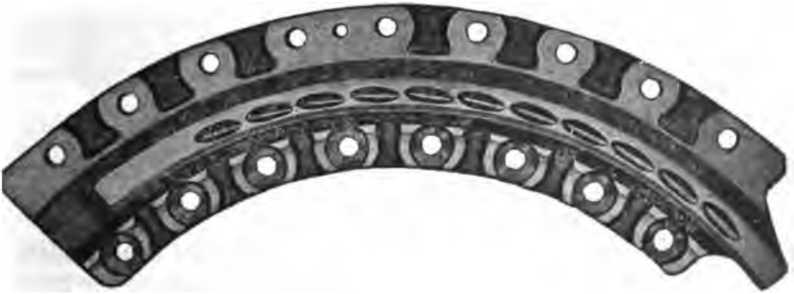
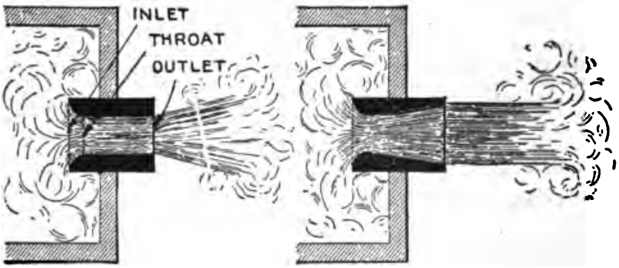


FIG. 2,378.—Curtis stationary nozzles for 300 kw. turbine.

If the blades were made as shown the area of the passages would not be uniform and there would be spaces as at M M′ not filled with steam. Hence, in practice, blades are made thicker at the center than at the ends.

Nozzles and Flow of Steam.—Since the action of a turbine depends on the motion of the steam, the correct shaping of the nozzles to give the proper flow is essential. There are two important types, the parallel nozzle, fig. 2,379, and the diverging nozzle, fig. 2,380. The three principle sections of a nozzle are:

1. The inlet;



FIGS. 2,379 and 2,380.—Parallel and diverging nozzles. The distinction between the two is that expansion of the steam takes place after discharge from a parallel nozzle, and before discharge in diverging nozzle. In the first case, the discharge takes the form of a conical jet and in the second, that of a cylinder. The velocity is greater with a diverging nozzle since it is not reduced at discharge by lateral expansion.

2. The throat;
3. The outlet.

Steam in flowing through a nozzle passes the inlet at the initial pressure; a definite fall of pressure takes place between the inlet and the throat, depending on the initial and outlet pressures and the type of nozzle; the pressure again falls, or remains constant between the throat and the outlet, depending on the type of nozzle and the pressure at the outlet.

According to Napier's experiments, the weight of steam discharged in a given time when flowing through a parallel nozzle having a rounded inlet, as in fig. 2,379, depends only on the absolute initial pressure, so long as the absolute pressure, against which the nozzle discharges, does not exceed .6 of that pressure. If, however, the final pressure be more than .6 of the initial pressure, the weight discharged will be less than before, and will become very much less as the pressure difference decreases.

Later tests show that the weight of steam discharged increases until the lower pressure drops to 58 per cent of the higher pressure; this is generally accepted as being correct. This is called the *critical pressure*, and is the minimum pressure reached at the throat, no matter how low the pressure may be at the outlet.

FIGS. 2,381 and 2,382.—Two views of Westinghouse nozzle chamber with stationary guide blade section attached.

When a parallel nozzle discharges into a medium having a pressure not greater than the critical pressure, the maximum velocity of discharge is about 1,450 feet per second. The steam, in passing through a parallel nozzle, does not expand between the throat and the outlet. In most turbines the steam flows through passages between guide vanes, which are so shaped as to virtually form groups of parallel nozzles—that is, passages of uniform cross section area, so that expansion does not take place within these passages.

Where velocities higher than 1,450 feet per second are required, or where the pressure at the nozzle outlet is lower than the critical pressure, a more efficient jet can be secured by using the diverging nozzle, as in fig. 2,380. In a nozzle of this type steam expands to the outlet pressure within the nozzle itself, hence, at the discharge, it takes the form of a solid cylindrical jet, having a diameter, as shown, equal to that at the outlet. It is evident that the velocity of discharge must be greater than with the parallel nozzle as, with the latter, the lateral expansion of the steam, on leaving the nozzle, considerably reduces the velocity.

To secure the maximum velocity with a diverging nozzle the taper of the walls must be correctly proportioned, to allow the steam to expand within the nozzle to the pressure at the outlet—that is, there must be *complete expansion*. In this way velocities as high as 3,000 to 4,000 feet per second may be obtained. These high velocities are necessary in the operation of simple impulse turbines, in order to avoid loss in converting the pressure energy of the steam into kinetic energy. Turbines of this type must then run at great tangential velocity, because for efficient working the peripheral speed of the wheel should be nearly one-half that of the steam.

Classification.—Turbines may be classified in several ways; a common, yet erroneous division being with respect to the action of the steam, as:

1. Impulse;
2. Reaction.

Both words, as used in this connection, are misnomers, and therefore misleading, as will later be explained.

With respect to whether there be one or more revolving discs, or *rotors*, in one *compartment*, turbines are called:

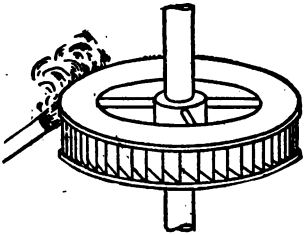
1. Simple;
2. Compound;

and with respect to the number of step reductions of pressure, they are classified as:

1. Single stage;
2. Multi-stage.

There are also several divisions depending on the direction in which the steam flows, as:

1. Radial { inward flow, outward flow;
2. Axial, or parallel flow;
3. Mixed flow.



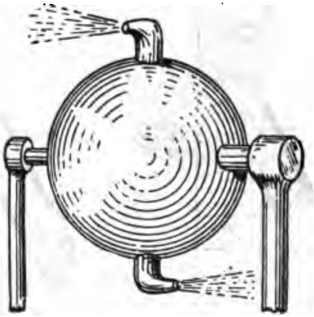


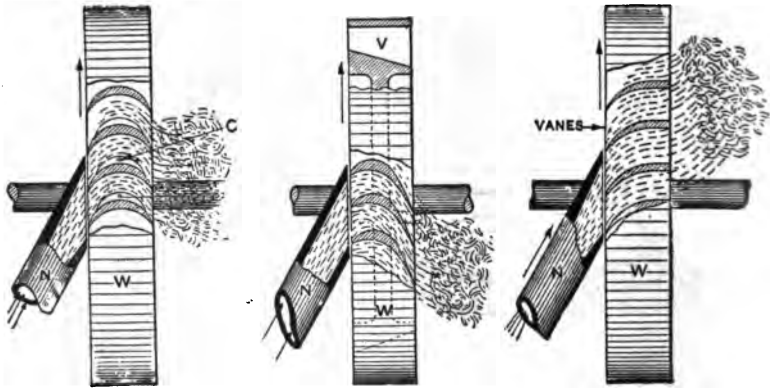
Fig. 2,383.—The first impulse turbine; invented by Giovanni Bianca, of Loreto, Italy, in the year 1629. *Pure impulse operation.*

Fig. 2,384.—Ancient reaction turbine, as described by Hero, of Alexandria, in the year 100 B. C. *Pure reaction operation.*

Impulse and Reaction.—Giovanni Bianca, of Loreto, Italy, in the year 1629 invented a turbine very much like a miniature water wheel, as shown in figure 2,383, and operated by a jet of steam; this represents a simple and pure *impulse* wheel.

In a book written by Hero, of Alexandria, in the year 100 B. C., a very simple form of pure reaction turbine is described. As shown in figure 2,384, it consisted of a hollow sphere, free to rotate on two trunnions, through one of which it received steam from a boiler. The sphere was provided with two opposite projecting nozzles, at right angles to the axis of the trunnions, and pointing in opposite tangential directions. The reaction of the steam escaping from the nozzles caused the sphere to revolve on its trunnions, in much the same way that water escaping from the arms of a lawn sprinkler causes it to revolve.

The two turbines just described are presented simply to illustrate the operation of a turbine by impulse and by reaction.



FIGS. 2,385 to 2,387.—Elementary turbines illustrating *impulse* and *reaction*. Fig. 2,385, so called impulse wheel; fig. 2,386, pure impulse wheel in which there is no reaction; fig. 2,387, so called reaction wheel.

All modern turbines, although grouped into two classes, as impulse and reaction, operate by the combination of these two forces.

Fig. 2,385 shows the usual type of blades in the so called impulse turbine, and fig. 2,386, the necessary shape for impulse alone, without any reaction. Fig. 2,387 shows what is known as a reaction turbine. By comparing figs. 2,386 and 2,387, it will be seen that the operation of these wheels is, in each case, due to both impulse and reaction. The distinction between the two types depends on the shape of the nozzle and vanes.

The use of the terms "impulse" and "reaction" is unfortunate, but since their misapplication has become general, it is necessary to understand the significance as applied to turbine operation.

In the so called impulse type, fig. 2,385, the nozzle is of a diverging cross section, allowing the steam to expand therein; it thus attains a high velocity, and impinges upon the moving vanes. The steam, in passing through the wheel, imparts some of its kinetic energy to the blades, and leaves them at a lower velocity, but *at the same pressure* at which it left the nozzle.

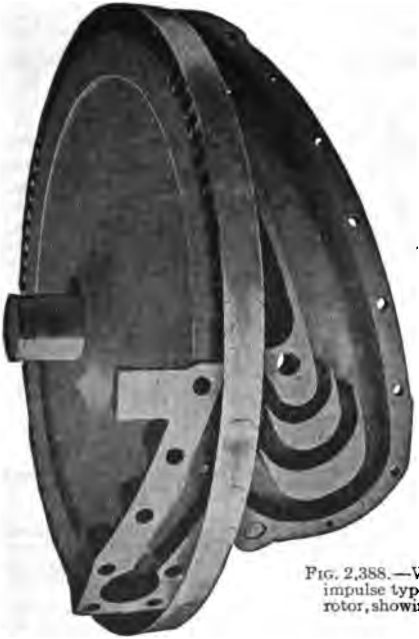
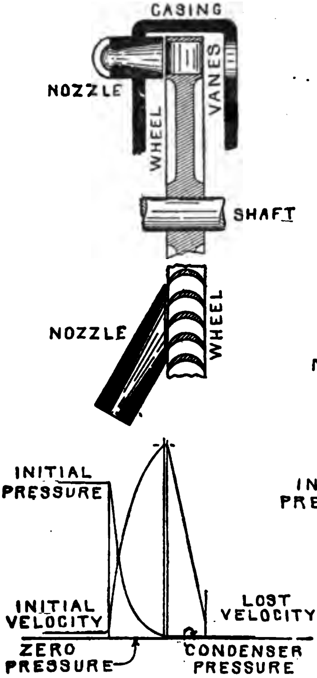
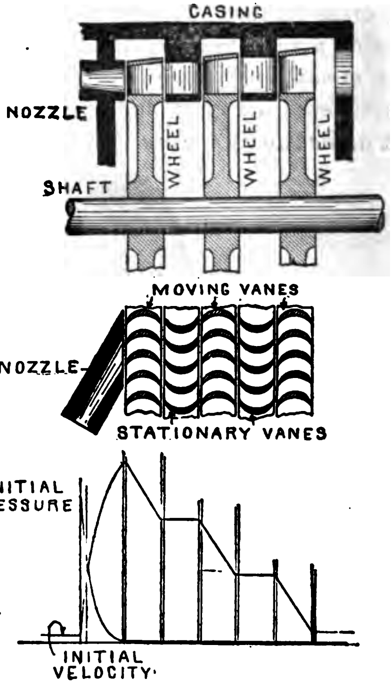


FIG. 2,388.—View of Westinghouse impulse type single wheel turbine rotor, showing nozzle and reverser.

In the so called reaction type, fig. 2,387, the nozzle has parallel sides, hence the steam passes through it at practically constant pressure, expansion taking place during its flow through the wheel. The passages are of diverging cross section to permit this expansion, *thus reducing the pressure.* It is important to note from the foregoing that:





FIGS. 2,389 to 2,391.—Simple impulse turbine and pressure velocity diagram. ***In operation,*** the steam is completely expanded in the nozzles, being reduced to condenser pressure at the outlet. The velocity falls in passing through the wheel, a certain amount of energy being lost by the inability of the wheel to absorb all the velocity generated in the nozzle.

FIGS. 2,392 to 2,394.—Compound impulse turbine and pressure velocity diagram. Steam, after leaving the nozzles, passes through several wheels separated by sets of stationary valves. As with the simple turbine, the steam is completely expanded within the nozzles, issuing therefrom at condenser pressure. The velocity generated in the nozzles is absorbed in steps as the steam flows through the wheel, and remains constant during the passage through the stationary vanes. The object of compounding is to reduce the speed of rotation.

In the **impulse type,** *steam enters and leaves the passages between the vanes at the same pressure.*

In the **reaction type,** *the pressure is less on the exit side of the vanes than on the entrance side.*

Expansion is confined to the nozzle in the impulse type, and to the vane passage in the reaction type.

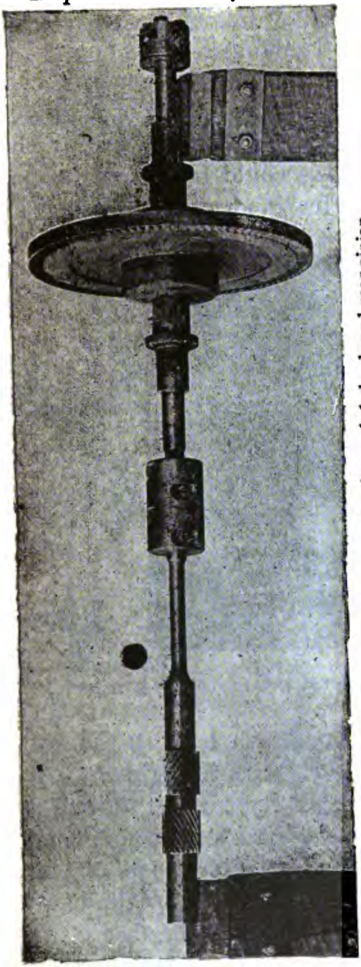


FIG. 2,395.—Westinghouse, impulse type single rotor, showing sealed glands and gear pinion.

Simple Impulse Turbine.—A simple turbine is one having only a single wheel. The principal features of this type are illustrated in figs. 2,389 to 2,391. It consists of a single wheel enclosed in a casing, or compartment, and having one or more expansion nozzles, which direct the steam at an acute angle onto the vanes. The diagram fig. 2,391 shows the action of the steam in passing through the turbine.

It should be noted that the steam pressure is at a maximum at the inlet of the nozzle, and at the outlet is reduced to the pressure of the condenser. The pressure then is the same on both sides of the wheel, hence there is no leakage of steam through the clearance spaces.

The velocity of the steam is greatly increased in passing through the expansion nozzle, reaching, under conditions shown in the diagram, a speed of about 4,000 feet per minute. The revolutions of the wheel must, therefore, be very high, and although the velocity is greatly lowered as the steam passes through the vane passages, it will leave the wheel with considerable residual velocity, which represents so much lost energy. There are other losses, such as friction in the bearings, friction of steam on wheel, etc.

Compound Impulse Turbines.—Part of the energy remaining in the steam as it leaves the vane passages of a single turbine could be converted into useful work by causing it to act on another wheel. The two wheels could be keyed to one shaft by interposing a set of stationary guide vanes, having a reverse curve, so as to deflect the steam on leaving the first wheel and cause it to enter the second wheel at the proper angle.

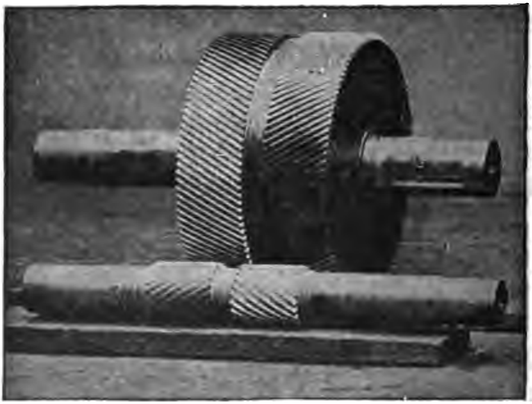


Fig. 2,396.—Westinghouse, impulse type single wheel turbine reduction gear and pinion; made from carbon steel forgings. The gears are of the split helical or herringbone type.

The speed of rotation, then, would not have to be so high as with a single wheel in order to absorb the same amount of energy from the steam. This arrangement is called a *compound turbine.*

In some cases more than two wheels are keyed to the shaft, the number depending on the revolutions, and the desired reduction in the velocity of the steam.

The application of the compound principle is illustrated in figs. 2,392 to 2,394. Here three wheels are shown keyed to the shaft, being separated by two sets of stationary vanes, all being enclosed on one compartment.

As in a simple turbine, steam enters the compartment through an expansion nozzle, so as to generate a high initial velocity; passing from the nozzle outlet it impinges upon the vanes of the first wheel, and then passes through the first set of stationary vanes, which are curved in a reverse direction to redirect the steam to the second wheel. It passes thence through the second set of stationary vanes to the third wheel, and on to the exhaust outlet.

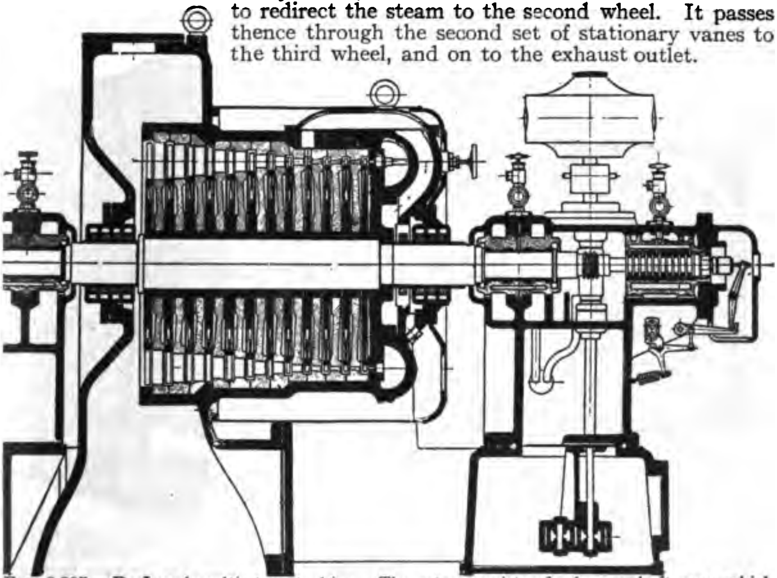
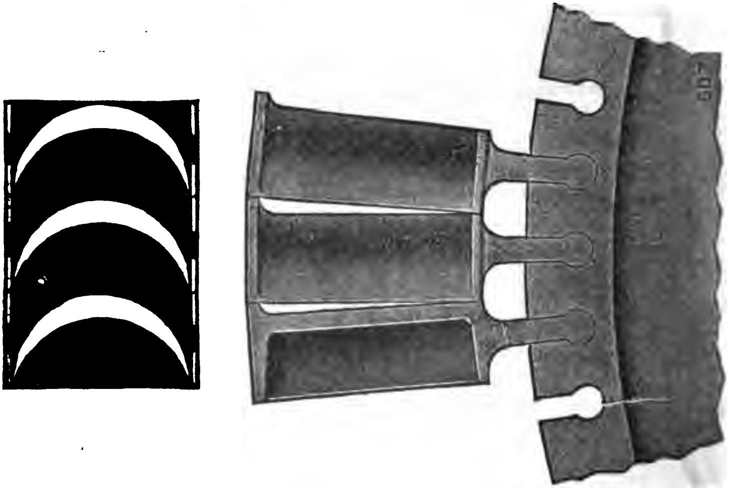


FIG. 2,397.—De Laval multi-stage turbine. The rotor consists of a heavy shaft upon which is mounted a series of discs or wheels, each revolving in an independent chamber formed between diaphragm held in a cylindrical casing. Steam is admitted to the steam chest at the right hand end of the casing, and flows thence through nozzles and impinges upon the buckets of the first wheel. The nozzles employed in the first stage are formed of tubes carefully bored and reamed and set in the nozzle ring, or they may be bored and reamed directly in the nozzle ring itself. The nozzles of this stage occupy only a portion of the circumference, to avoid the difficulties of very short blade length always encountered where full admission is employed in the first stage. Any or all of these nozzles may be controlled by hand operated valves seating upon the inlet openings. These valves, however, are not used for speed regulation, and are not operated automatically, as it has been found that the opening or closing of one or more nozzles at a time by a governor is apt to induce surgings and fluctuations in speed, especially where several turbo-generators are operating in parallel or supplying current to rotary converters. The purpose of hand control of the several nozzles of the De Laval turbines is to permit of the admission of more or less steam as the load changes greatly at different periods.

The velocity of the steam, as indicated in the diagram, falls during its passage through the wheels, but remains constant in passing through the stationary vanes. Since the velocity is gradually decreased as it passes through the several wheels, it must be evident that if the same *quantity* of steam is to flow through successive wheels in the same interval of time, the passages must gradually increase in size.

The passages through the stationary vanes are of uniform cross section, because the velocity here is constant, however, it should be noted that the second set of stationary vanes is larger than the first, since the velocity of the steam is less at the exit of the second wheel than at the first.



Figs. 2,398 and 2,399.—Method of fastening De Laval buckets in rim of turbine wheel. The buckets made of a nickel and copper alloy are drop forged. The tips of the buckets have projections or lugs which fit against similar projections on the adjacent buckets, forming a continuous rim, which is advantageous in diminishing the fan action of the buckets and also in that it prevents spilling by confining the jets of steam within closed channels. The rim also maintains the proper spacing. The buckets are secured to the rim by transverse dovetails. The increase in the cross sectional area of the passages required by the expansion of the steam as it proceeds through the turbine is gained by lengthening the blades, reducing the diameters of the wheels correspondingly and increasing the bore of the casing. The length and strength of the blades used in the last wheel really determine the maximum speed at which the turbine may be operated, while the length of these buckets and the diameter of the wheel determine the maximum capacity for given steam conditions. A proper balance of these factors, and proportioning of the last wheel with respect to diameter, pressure drop, capacity and terminal pressure, is rendered possible by the use of a speed reduction gear. The wheels upon which the buckets are mounted consist of hydraulically forged steel discs, finished and ground on all surfaces. The hub of each wheel is so extended that the hubs of adjacent wheels touch one another, permitting them to be locked in place by one nut, and at the same time adding considerably to the stiffness of the shaft. The wheels are mounted upon the shaft by means of taper bushings.

Simple Multi-Stage Impulse Turbines.—In the simple turbine, figs. 2,389 to 2,391, it will be noted that the boiler pressure is reduced to exhaust pressure in one step; the excessively high steam velocity thus generated calls for a very high number of revolutions of the wheel to efficiently absorb the energy of the steam. If the speed of the wheel be reduced, there would remain considerable residual velocity of the steam after leaving the vane passages, resulting in undue loss. It has been shown that this lost energy could be absorbed by a second wheel working compound, as in figs. 2,392 to 2,394, thus permitting slow speed, without loss in economy.

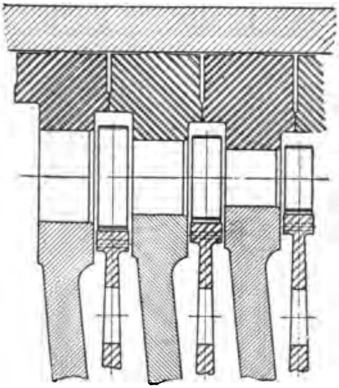
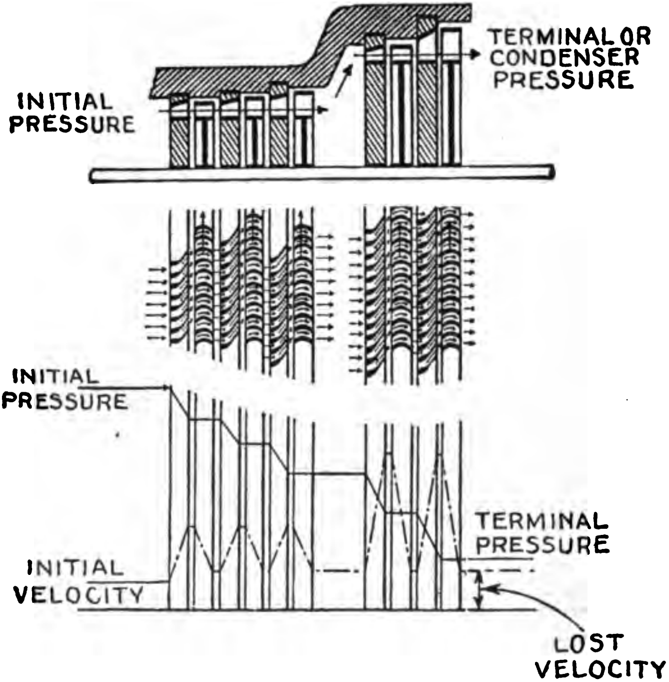


FIG. 2,400.—Partial section of De Laval multi-stage turbine showing the steel retaining rings which form a continuous armor enclosing the wheels. With the exception of the nozzles in the first stage, the nozzles of each succeeding stage are formed between nickel bronze guide vanes set around the entire periphery of the diaphragm. The guide vanes are spaced and located upon the rim of the diaphragm by pins and are held in place by a solid steel band shrunk over their tips. Two pins are used for each vane to determine its proper angle, and therefore in connection with the shape of the vanes, to fix the contour and cross sectional area of the nozzles formed between the successive vanes.

A turbine may be designed to run at slow speed without loss by employing the *multi-stage* method of transforming the pressure energy of the steam into kinetic energy. The word "stage" as applied to turbines, relates to the conditions of pressure under which this transformation takes place.

The application is much the same as when applied to the steam engine. Thus, a triple expansion engine is sometimes referred to as a three stage expansion engine, that is, the expansion of the steam in passing through the engine is divided into three steps, part of it taking place at high pressure, part at intermediate pressure, and part at low pressure, there being a separate cylinder for each step.



Figs. 2,401 to 2,403.—Simple multi-stage impulse turbine and pressure velocity diagram. The word "stage" relates to working pressure. In a multi-stage turbine, steam, in passing through the turbine, works at several pressures. There is a separate compartment and set of nozzle for each stage; the pressure of the steam is constant in passing through a compartment, but drops successively as it passes through the several sets of nozzles, regenerating velocity at each reduction of pressure. Modern multi-stage turbines sometimes have as many as twenty to thirty stages.

A multi-stage turbine, then, is one in which the pressure energy of the steam is progressively transformed into kinetic energy in two or more pressure stages. In construction there is a separate set of nozzles, and a separate compartment for each stage, as shown in figs. 814 and 815.

In operation, since there is only one wheel in each compartment, it is evident that for efficient working the velocity of the entering steam must be proportioned to the speed of the wheel, and must be such that it will be reduced to the exhaust velocity in passing through the vane passages. For, since the entrance velocity depends on the amount of pressure reduction at the nozzles, any excess velocity after leaving the wheel represents an undue pressure loss, which otherwise would be available for generating velocity in the second set of nozzles for the second wheel. Now as the wheels run at slow speed the entrance velocity should be moderate to avoid the loss just mentioned. Hence, with the proper number of stage, the reduction of pressure at each set of nozzle may be so proportioned as to give the correct entrance velocity to avoid loss.

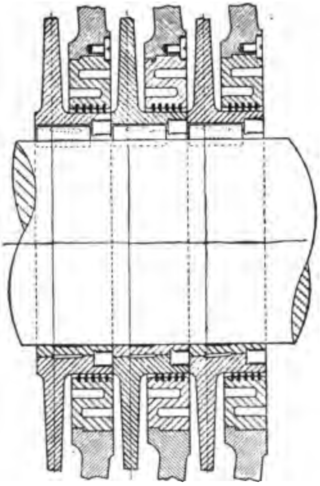


FIG. 2,404.—Method of attaching wheels to De Laval turbine shaft by means of tapered sleeves, which are drawn into place by a nut and are held from rotating on the shaft by keys. The labyrinth packing between the diaphragms and the hubs of the wheels is also shown.

The relations of pressures and velocities for the several stages are clearly shown in the diagram. It should be noted that the velocity falls to that of the exhaust during each stage, being regenerated in passing through each set of nozzle by successive reductions of pressure.

The multi-stage turbine is subject to leakage at points where the shaft passes through the walls of the compartments, as at A B, which requires

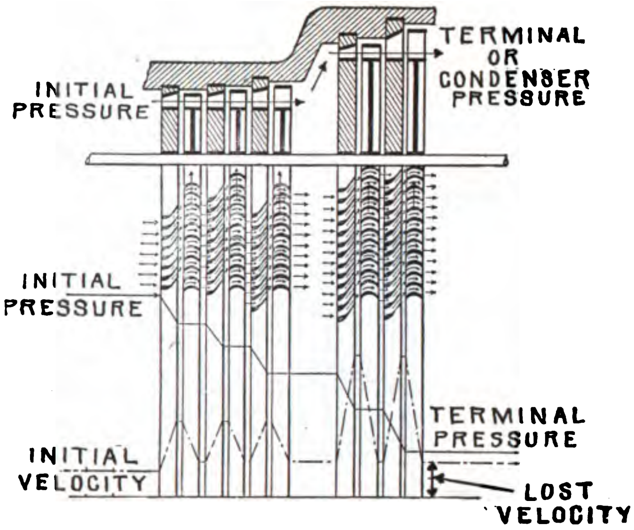
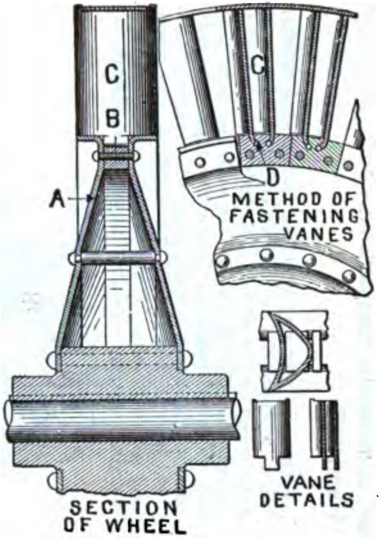
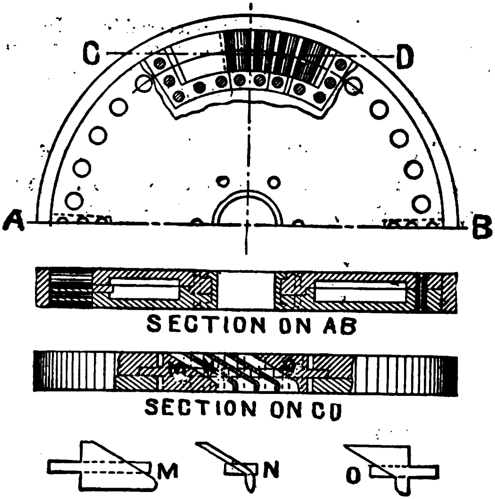


FIG. 2,405.—Diagram of Hamilton-Holzworth simple multi-stage impulse turbine, showing course and action of the steam.



FIGS. 2,406 to 2,410.—Hamilton-Holzworth turbine wheel construction. Conical discs A, are riveted to a cast steel hub. There is a space between the discs at their periphery, in which are riveted short steel segments B. The vanes are attached to these segments, as shown in the sectional view at D. The vanes are formed with lips extending downward, and these lips are enlarged at their ends to fit into the enlarged bottom portions of cross channels in segments B. The vanes are so designed that passages through the wheels have a uniform area from beginning to end, and they are made hollow to reduce their weight. On the outer ends of the vanes a thin steel band is shrunk to give an outside wall to the steam channels. The vanes are milled on both edges to give correct influx and efflux angles.



FIGS. 2,411 to 2,416.—Hamilton-Holzworth stationary discs. These are built up of two side pieces riveted together. Each vane is a separate piece held by a projection at its lower end, which fits in an annular groove between the two discs at their periphery. The vanes are of drop forged steel and are secured by rivets. After they are in position, their outside ends are ground and a steel ring is shrunk on. In figs. 2,414 to 2,416, M, N, O, are three forms in which the drop forged vanes may be made. In the turbine, as now constructed, however, the vanes extend around the whole periphery, so that the type shown at 2,415 is the only one required

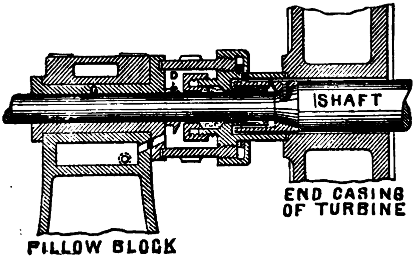
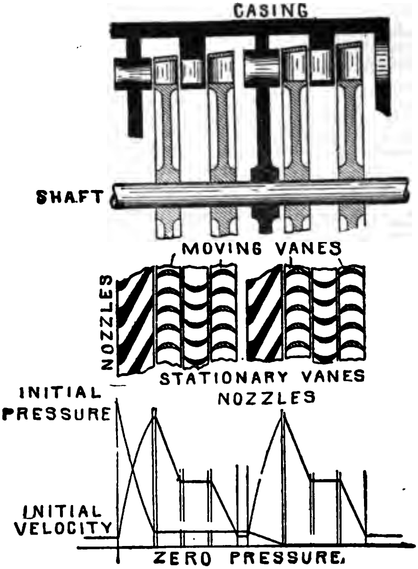


FIG. 2,417.—Hamilton-Holzwarth gland and shaft bearing; sectional view at the point where it passes through the end of the turbine casing. The shaft is turned to a smaller diameter at its end, and runs in a bushing G, having a flange bearing against the inner side of the pillow block. At A, is a cylindrical piece attached to and rotating with the shaft. This piece projects into an annular groove in the piece B, but it does not completely fill the groove and a circuitous passage is formed, through which the steam must pass before reaching the stuffing box C. The object of this passage is to provide condensing surface so the steam itself will not reach the packing. The joint at the stuffing box is thus practically a water locked joint. To prevent the oil working into the turbine, a bushing F, is attached to the shaft, which throws off the oil into the space D, by centrifugal force, where it drips down through a channel E, into a compartment in the pillow block. Any water escaping through the stuffing box is also collected in the same compartment. The bearing is oiled by a forced oil system, the oil being supplied to the bottom of the bushing.

special packing, and, unfortunately, is inaccessible. However, multi-stage turbines are built with a large number of compartments, frequently thirty or more, hence the successive drops in pressure are very small, so there is little tendency to leakage through stuffing boxes.

Compound Multi-Stage Impulse Turbine.—In order to reduce the large number of stage necessary in the simple multi-stage turbine, especially when working at high pressures, the compound principle has been applied by placing two or more wheels in each compartment. Thus a greater pressure reduction may be made between stages, and in this way the number reduced.



FIGS. 2,418 to 2,420.—Compound multi-stage impulse turbine and pressure velocity diagram. It is virtually two or more compound turbines joined in series, or the equivalent, a multi-stage turbine having more than one wheel in each compartment. The object of compounding is to reduce the number of stages, especially in case of high initial pressure.

A compound multi-stage impulse turbine then, *is one in which* *the* **pressure energy** *of the steam is progressively transformed into* **kinetic energy** *in two or more stages, with compound working in each stage.* In construction there are two or more compartments, each containing two or more wheels, each compartment being connected with the next by a set of nozzles, as shown in figs. 2,418 to 2,420.

The illustration shows only two compartments, each having two wheels; however, there may be a larger number of compartments, or wheels, depending upon conditions. The action of the steam in passing through the turbine is shown in the diaphragms. Higher steam velocities may be employed, thus permitting a greater pressure reduction between stages, and decreasing the number necessary. This is the principle of the compound multi-stage turbine.

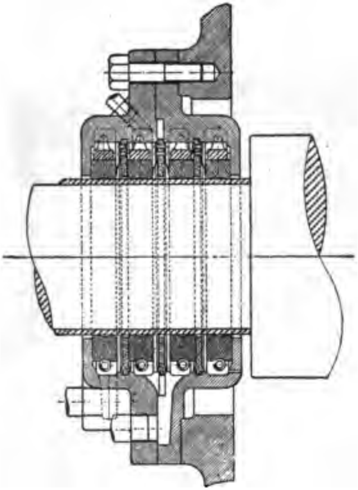
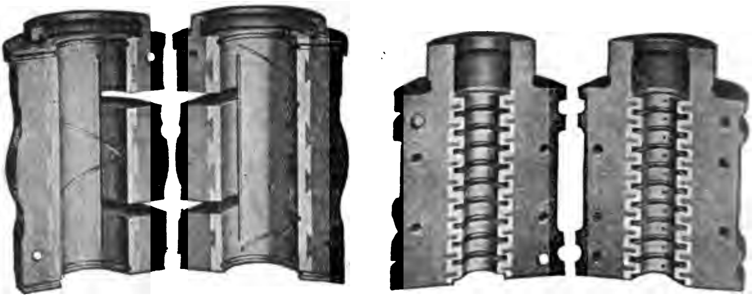


FIG. 2,421.—Carbon packing at low pressure end of De Laval multi-stage turbine. Provision is made for introducing live steam at reduced pressure between the second and third rings, so that any leakage into the turbine will be of steam, not air.

A compound multi-stage impulse turbine, then, is one which has two or more compartments, each containing two or more wheels, and a set of nozzles.

The application is shown in figs. 2,418 to 2,420. The illustration shows two compartments, each provided with a set of nozzle. Each compartment contains two wheels and a set of stationary vanes; in construction there may be more wheels in each compartment, or more compartments, depending on conditions. It will be noted from the diagram that the pressure remains constant in each compartment, being reduced as the steam flows through the nozzles, also, the velocity falls in two steps in each compartment, remains constant in the stationary vanes, and rises in the nozzles.



FIGS. 2,422 to 2,425.—Main turbine and thrust bearings for De Laval multi-stage turbine. Both bearings are split to permit their removal without disturbing the shaft.

Reaction Turbines.—Usually, turbines of this type are built compound, as shown in figs. 2,429 to 2,431; the principal parts consist of numerous rows of moving vanes, separated by alternate rows of fixed guide vanes. The passages through the latter are of increasing cross section, hence part of the expansion takes part therein, being completed in the wheel. It should be noted that the fixed guides perform the functions of nozzles, and the steam in passing through the turbine expands continually from boiler pressure to exhaust pressure.

The diagram indicates the increase of velocity as the steam flows through

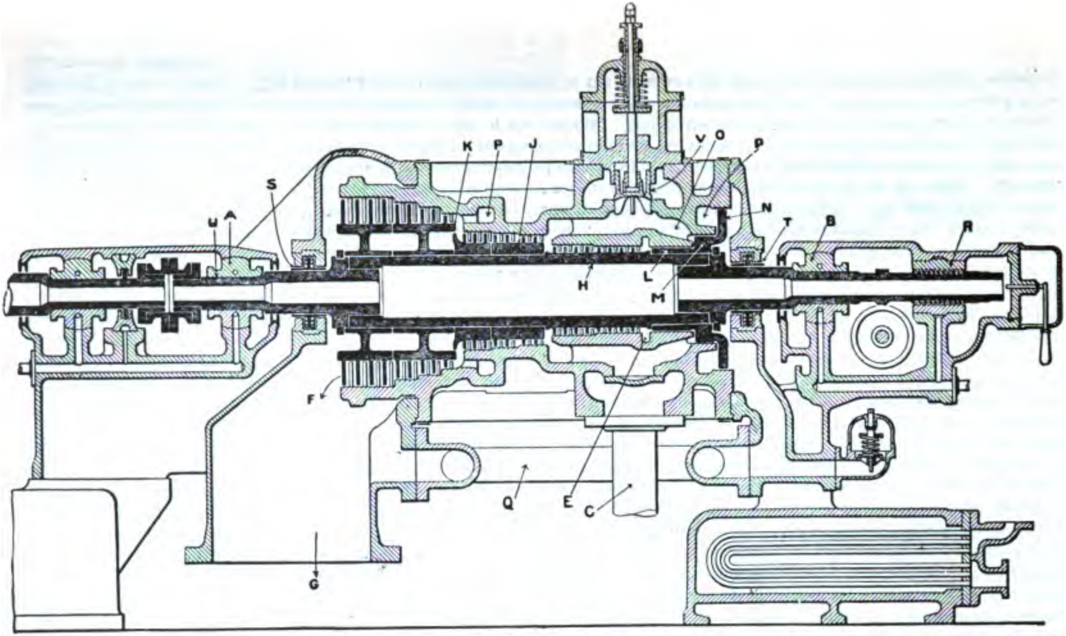


FIG. 2,426.—Sectional view of Allis-Chalmers Parsons or reaction type turbine. The principles underlying the various styles of Allis-Chalmers steam turbines referred to are essentially the same, and while there are necessarily differences in the design and construction, the same description in general applies to all of them. In the figure all details which are not necessary to an understanding of the general principles are omitted. The turbine consists essentially of a fixed casing, or cylinder, and a revolving spindle, or drum. The ends of the spindle are extended in the form of a shaft, carried in two bearings A and B, and excepting the small parts of the governing mechanism and the oil pump, these bearings are the only rubbing parts in the entire turbine. Steam enters from the steam pipe at C, after passing through the main regulating valve D. This valve is operated by the governor through suitable controlling mechanism. The steam enters the cylinder through the passage E, and, turning to

FIG. 2,426.—*Text continued.*

the left, as seen in the cut, passes through alternate stationary and revolving rows of blades, finally emerging from them at F, and flowing through the connection G, to the condenser or to the atmosphere, depending upon whether the turbine is designed for condensing or non-condensing operation. Each row of blades, both stationary and revolving, extends completely around the turbine and the steam flows through the full annulus between the spindle and the cylinder. In an ideal turbine the lengths of the blades and the diameter of the spindle which carries them would continuously and gradually increase from the steam inlet to the exhaust. Practically, however, the desired effect is produced by making the spindle in steps, there generally being three such steps or stages, H, J and K. The blades in each step are arranged in groups of increasing length. At the beginning of each of the larger steps the blades are usually shorter than at the end of the preceding smaller step, the change being made in such a way that the correct relation of blade length to spindle diameter is secured. The details of the arrangement and construction of the blades will be explained later. The steam, acting as previously described, produces a thrust tending to force the spindle toward the left, as seen in the cut. This thrust, however, is counter-acted by the "balance pistons" L, M and N, which are of the necessary diameter to neutralize the thrust on the spindle steps, H, J and K, respectively. These elements are called "pistons" for convenience, although they do not come in contact with the cylinder, but both the pistons and the cylinder are provided with alternate rings which form a labyrinth packing to retard the leakage of steam. In order that each balance piston may have the proper pressure on both sides, equalizing passages O, P and Q, are provided, connecting the balance pistons with the corresponding stages of the blading. The end thrust being thus practically neutralized by means of the balance pistons, the spindle "floats" so that it can be easily moved in one direction or the other. In order to definitely fix the position of the spindle, a small adjustable collar bearing, is provided at R, inside the housing of the main bearing B. This collar bearing is adjustable so as to locate and hold the spindle in such position that there will be such a clearance between the rings of the balance piston and those of the cylinder, that the leakage of steam will be reduced to a minimum and, at the same time, prevent actual contact under varying conditions of temperature. When the shaft passes out of the cylinder at S and T, it is necessary to provide against in leakage of air or out leakage of steam. This is accomplished by means of glands as shown in figs. 2,463 to 2,468. These glands are practically frictionless, being made tight by water packing without metallic contact. The shaft of the turbine is extended at U, and coupled to the shaft of the alternator by means of a flexible coupling as shown in fig. 2,462. The high pressure turbines are so proportioned that, when using steam as previously described, they have enough capacity to take care of the ordinary fluctuations of load when controlled by the governor through the valve D. To provide for overloads, the valve V, is supplied to admit steam to an intermediate stage of the turbine. This is the equivalent of the by pass valve sometimes used for admitting live steam to the low pressure cylinder of a compound reciprocating engine. This valve is arranged to be operated by the governor. The foregoing is a general description of the Parsons type of turbine, and of the elements necessary to successful operation.

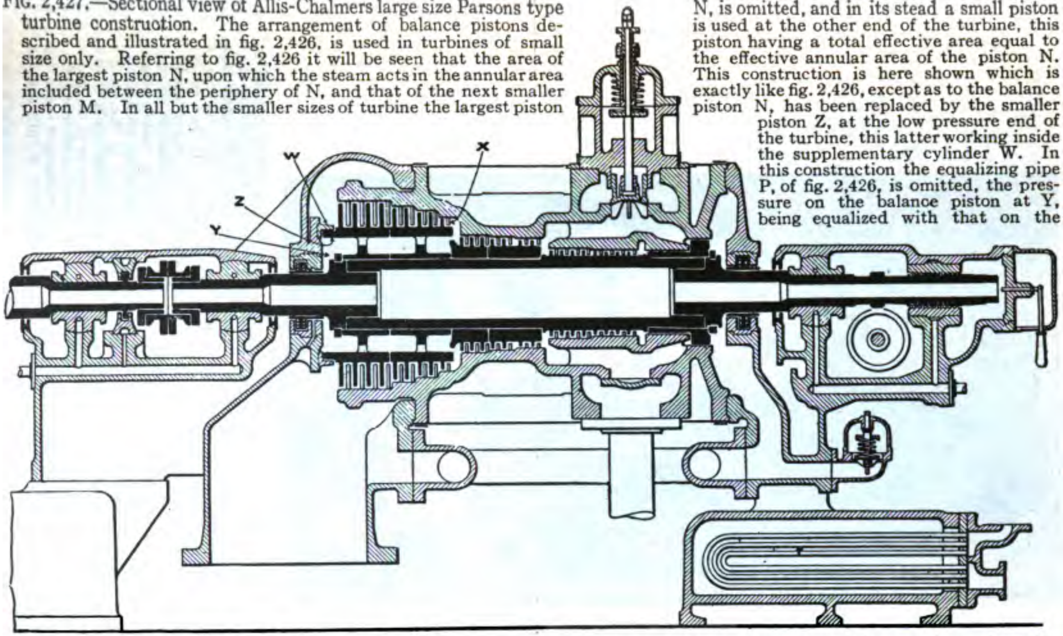


FIG. 2,427.—Sectional view of Allis-Chalmers large size Parsons type turbine construction. The arrangement of balance pistons described and illustrated in fig. 2,426, is used in turbines of small size only. Referring to fig. 2,426 it will be seen that the area of the largest piston N, upon which the steam acts in the annular area included between the periphery of N, and that of the next smaller piston M. In all but the smaller sizes of turbine the largest piston N, is omitted, and in its stead a small piston is used at the other end of the turbine, this piston having a total effective area equal to the effective annular area of the piston N. This construction is here shown which is exactly like fig. 2,426, except as to the balance piston N, has been replaced by the smaller piston Z, at the low pressure end of the turbine, this latter working inside the supplementary cylinder W. In this construction the equalizing pipe P, of fig. 2,426, is omitted, the pressure on the balance piston at Y, being equalized with that on the third stage of the blading at X, by means of passages through the body of the spindle. This piston Z, is not only much smaller than the equivalent piston N, in fig. 2,426, but is very much stiffer, being backed up by the body of the spindle. As, due to varying temperatures, there is an appreciable difference in the axial expansion of the spindle and cylinder, and baffling rings in this construction of low pressure balance piston are so made as to allow for this difference. The high pressure end of the spindle being held by the collar bearing, the difference in expansion manifests itself at the low pressure end. The labyrinth packing of the high pressure and intermediate pistons has a small axial and large radial clearance, the same as in the construction of fig. 2,426, whereas the labyrinth packing of the piston at the low pressure end has a small radial and large axial clearance.

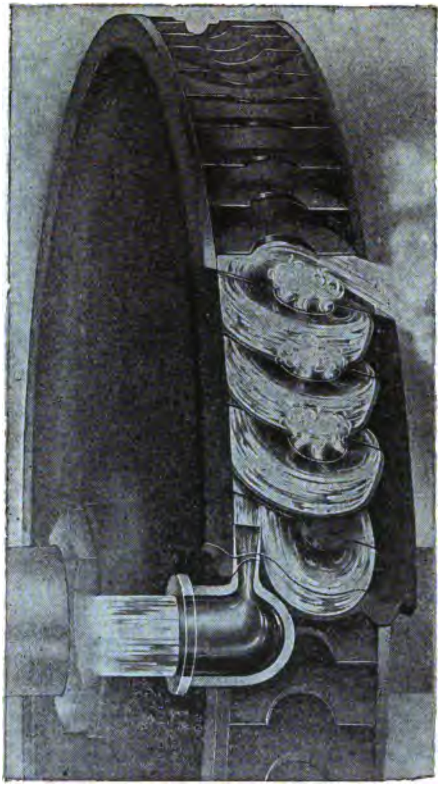
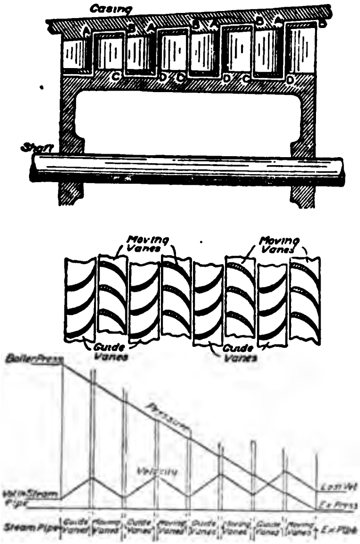


FIG. 2,428.—Terry turbine, view of wheel and reversing chambers showing operation. ***In construction,*** the wheel is made from a single forging of special composition steel and the semi-circular buckets or pockets are milled from the solid metal. ***In operation,*** the jet of steam issuing from the nozzle, gives up part of its velocity in its first reversal in the wheel buckets, but still retains a considerable portion of its energy. It then passes to a reversing chamber, which returns it to the wheel buckets again. This is repeated several times until all of its energy has been given up to the wheel. This principle of operation which returns the steam several times to the *same row of moving buckets*, makes possible the efficient use of the steam in a single wheel with internal buckets. The power producing action of the steam in the wheel buckets takes place on the curved surfaces at the back of the bucket. As the only function of the blades is to split the steam jet, close blade clearance is not necessary, and wear of blades is of little consequence. The turbine is of the radial flow type, so that there is no end thrust. Nozzles are made separate from the reversing chambers so that the turbine can be adapted to give maximum economy under changed operating conditions, such as increased boiler pressure. To make such a change it is only necessary to insert new nozzles.

the guide vanes, and the decrease in passing through the adjacent moving vanes. The pressure falls gradually from the inlet to the outlet; it is, therefore, maintained higher in the turbine passages than the exhaust pressure, this being characteristic of the reaction principle.

Turbine Governors.—There are several methods of regulating the steam supply so that a uniform speed may be maintained under a fluctuating load. These may be classified as:



FIGS. 2,429 to 2,431.—Compound reaction turbine. In this type steam expands in passing through the moving vanes instead of in the nozzle, as with the impulse turbine. The pressure falls gradually from inlet to outlet.

1. Throttling;
2. Partial admission;
3. Intermittent admission;
4. By pass.

As in steam engine practice, governing by throttling is not an

economical method, but has the advantage of simplicity. A governor of this kind is shown in fig. 2,432.

The two weights B, are pivoted on knife edges A, with hardened pins C, bearing on the spring seat D. The governor body E, is fitted in the end of the gear wheel shaft K, and has seats milled for the knife edges. A spindle G, of small diameter passes between the weights, and has at its outer end an adjustment nut I, for regulating the tension of the spring which controls the speed of the turbine.

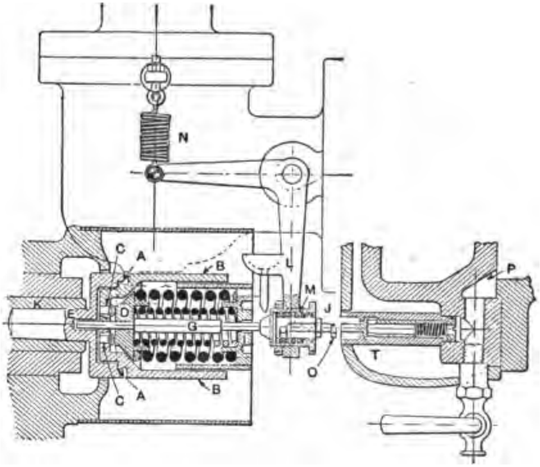


FIG. 2,432.—De Laval Throttling Governor. It is held in the end of one of the gear shafts by the taper plug K, and is made cylindrical in form, with its outer shell B, B, cut longitudinally into two halves which form the governor weights. These weights are fulcrumed at A, A, and have pins C, C, which press against a collar D, the latter taking the thrust of the spiral springs located within the governor. The movement of the governor is transmitted through the center spindle G, to the bell crank lever L, which is balanced by a spiral spring N. The shaft supporting this lever passes through the valve casting, on the inside of which are a pair of arms connecting with a double seated throttle valve as shown. A connection between the center spindle G, of the governor and the bell crank lever L, is a flexible connection, and that at the right, is a valve T, which connects through the passage P, with the wheel casing. If the speed become excessive, pin O, would strike spindle T, which would admit air to the wheel compartment and thus reduce the speed.

In operation, centrifugal force causes the weights to spread apart and press on the spring seat D; this pushes the governor pin G, forward, throttling the flow of steam to a degree corresponding to the load.

If the turbine be run condensing, a vacuum governor must be used, because the throttling governor will not control the speed within desirable limits for sudden fluctuations of load. The function of the vacuum valve is as follows; The governor pin G, actuates the plunger H, screwed into the bell crank L, however, without moving the plunger relative to the crank. This is on account of the spring M, being stiffer than the spring N, whose duty it is to keep the governor valve open and the plunger H, in contact with the spindle G. When a large part of the load is suddenly thrown off, the governor opens, pushing the bell crank in the direction of the vacuum valve T; this closes the governor valve, which is completely shut off when the bell crank is pushed so far forward that the screw O, barely touches the valve stem J. If this be not sufficient to check the speed, the plunger

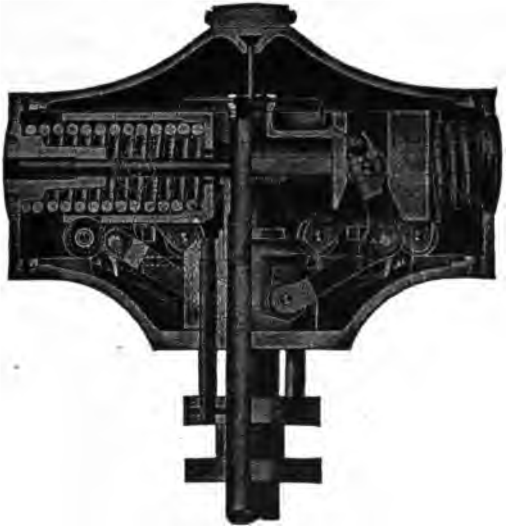


FIG. 2,433.—Interior view of Jahns type speed governor used with De Laval multi-stage steam turbine. It is mounted upon a vertical shaft driven from the turbine shaft by worm gearing. The weights act directly upon the springs in such a manner that the spring pressure is not transmitted to the governor mechanism. The weight moves in a horizontal plane. The movement of the sliding sleeve on the spindle is effected by bell cranks engaging both the weights and the sleeve through roller bearing connections, with races set at such angles that the movement of the weights due to centrifugal force is transmitted in such a manner that the force exerted by the sleeve is practically constant at each position of its travel. The oiling of the upper pins and slides within the governor casing is effected by an oil cup on top, which can be filled while in motion. The lower pins and slides move in a bath of oil, and any overflow oil is utilized in lubricating the spindle sleeve. The governing weight, springs and the entire mechanism are protected from dust and moisture by the enveloping case. The movement of the sleeve operates the governor valve through a simple system of links and levers, containing a spring link which prevents any injury to the governor valve through overtravel of the governor.

H, is pushed forward in the now stationary bell crank, and opens the vacuum valve. This allows the air to reach into space D, where the turbine wheel revolves, effectually checking the speed.

The partial admission method of governing regulates the steam supply by varying the number of open nozzles, an independent valve being provided for each nozzle. The valves are arranged to open successively, usually two-thirds of them being open at full load. The action of the regulating mechanism is such that the valves are either fully opened, or closed.

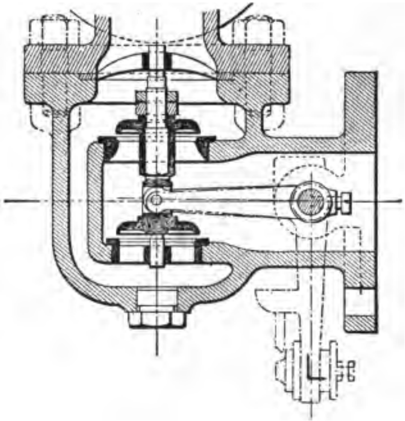


FIG. 2,434.—De Laval throttle valve; sectional view showing valve and seat. L, is the bell crank lever connection with the governor.

The power required to operate these valves is too great for direct movement by the governor, hence the governor controls their movements indirectly by setting in motion some mechanism powerful enough to do the work. The latter may be either electrical, mechanical or hydraulic. One construction of a partial admission governor is shown in fig. 2,435.

The action of the governor depends on the balance between the forces exerted by springs and the centrifugal forces of revolving weights.

The governor is supported by a flange keyed to the top of the vertical shaft of the turbine; and the whole supporting framework revolves with the shaft. The revolving weights A, are fulcrumed at the knife edges B; as the speed of the turbine increases, the centrifugal forces of the weights pull down the rod C, against the action of a heavy spring D. At E, a ball bearing joint forms a junction point between the revolving mass of the turbine and the stationary lever of the governor arm. An auxiliary spring F, is provided for varying the speed when synchronizing. The adjustment of this spring in large machines can be made from the switchboard by means of the small motor G.

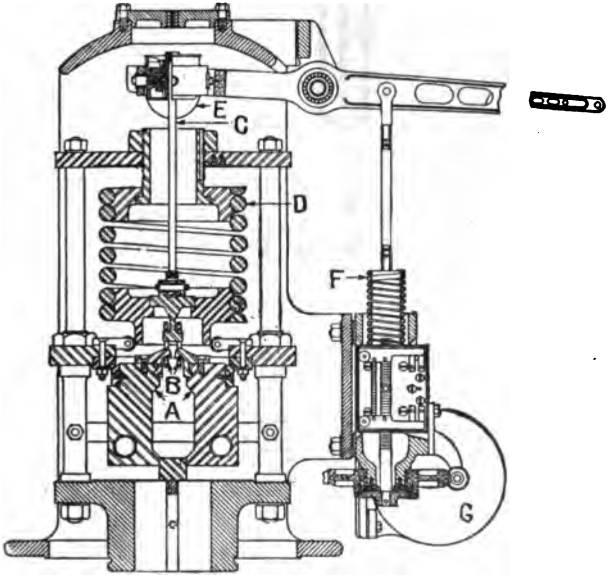
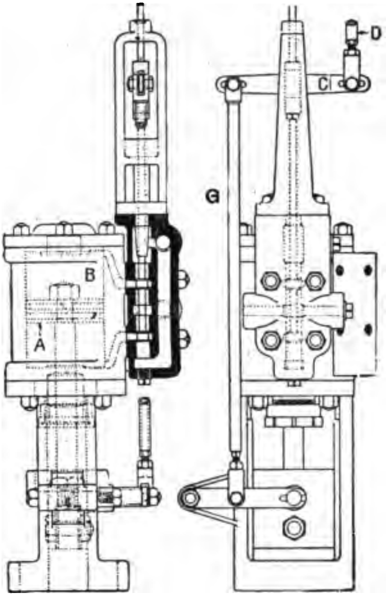


Fig. 2,435.—Curtis partial admission governor for vertical type turbines. The revolving weights A, fulcrumed at C, act by centrifugal force to pull down the rod C, against the action of the heavy spring D. Connection with the stationary lever is through the ball bearing gimbol joint E. An adjustable auxiliary spring F, permits varying the speed when synchronizing.

The movement of the governor lever is transmitted through the rod D, fig. 2,437, to the arm G, and to the pilot valve of the oil cylinder B, containing the piston A (fig. 2,436), which operates the main arm. The latter transmits the motion, either by means of a rack connecting with a pinion or by means of cranks to the rod carrying the cams. These cams act directly on the valves, opening and closing the number called for by the condition of the load.



Figs. 2,436 and 2,437.—Curtis hydraulic operating mechanism for valves. The movement of the governor lever is transmitted through the rod D, to the arm G, and to the pilot valve of the oil cylinder containing the piston A, which operates the main arm.

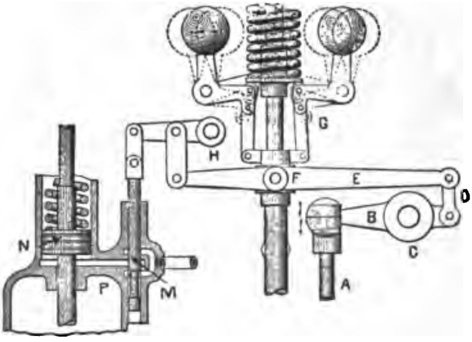


Fig. 2,438.—Westinghouse-Parsons intermittent governor; it is of the centrifugal type with bell crank levers, the vertical arms of which carry the balls, and the horizontal arms bear against the spiral spring which resists the centrifugal force of the balls. The tension of the spring may be adjusted for synchronizing. The details are fully described in the accompanying text.

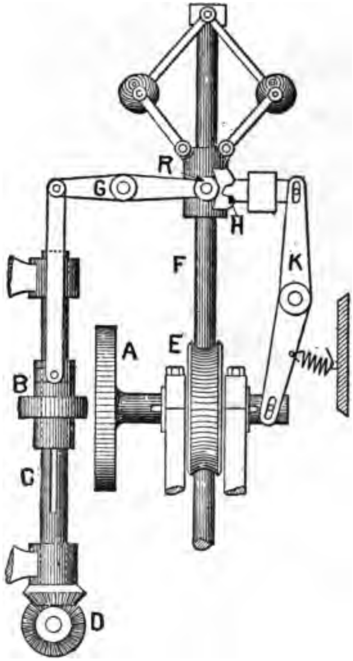
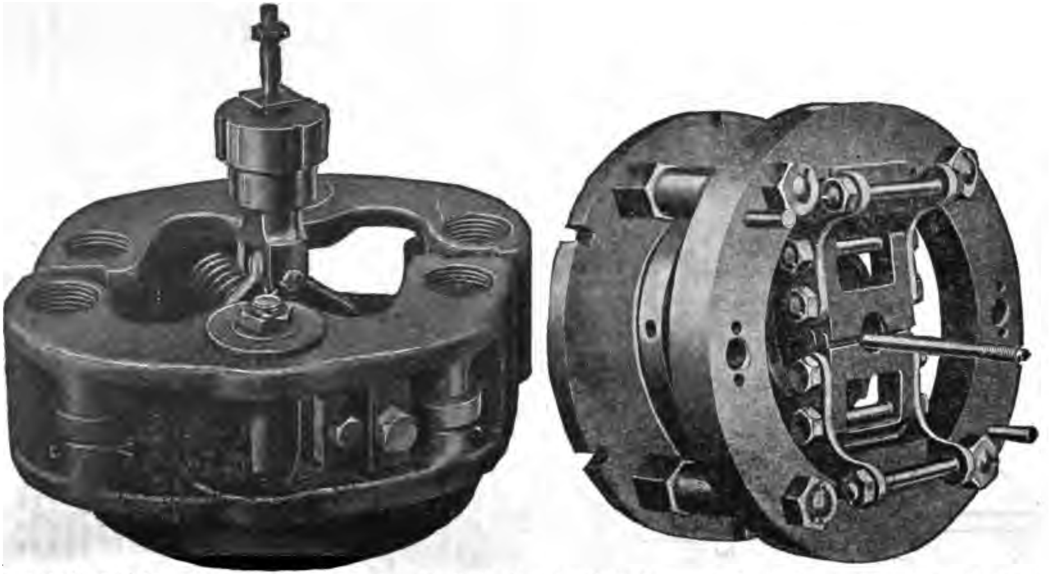


Fig. 2,439.—Hamilton-Holzworth turbine governor. Regulation is by throttling. The valve is controlled by means of a friction disc drive, in which the edge of a small disc or roller is in contact with the face of a continually rotating disc. This disc is driven by a worm and worm wheel from the shaft which operates the governor. At normal speed, the governor speed is in mid position and roller B, is at the center of the disc. If the turbine speed up, however, the governor sleeve will rise, carrying with it the right hand arm of lever G, which will move the roller B, to a corresponding distance downward. At the same time, the cam H, will be thrown to the right by contact with the roller R, upon the governor sleeve, and this, by means of a lever K, will move the disc A, and its shaft to the left in contact with roller B, thus imparting a rotary motion to the shaft C, and closing the throttle valve the necessary amount to bring the speed of the turbine back to normal again. For increase of load, the governor acts in a similar manner to admit more steam.

Intermittent admission governors are usually fitted to turbines of the reaction type. In this method of governing, steam is admitted to the turbines in puffs, the duration of each puff being dependent upon the load under which the turbine is working. For a light load the puffs are very short, thus admitting little steam, while for a heavy load, the puffs are longer, and consequently a greater amount of steam is admitted. An example of the intermittent admission type of governor is shown in fig. 2,438.

In construction, A, is a rod, connected to an eccentric which derives its motion from the main shaft of the turbine. It is attached by a ball joint to the lever B, fulcrumed at C. The link D, connects B, with another lever E, whose fulcrum F, may be shifted up or down by the governor G. The lever E, by means of other links and the fixed fulcrum H, is connected with the small relay valve M, whose motion from the midposition will admit steam under the piston N, which is directly attached to the steam of the admission valve. It is evident that the reciprocating motion of the rod A, due to the eccentric, will be transmitted to the valve M, causing the port



Figs. 2,440 and 2,441.—Curtis turbine governors. Fig. 2,440, Centrifugal shaft governor for 35 kw. machine; fig. 2,441, inertia governor. In the small machines the speed is controlled by a centrifugal governor mounted on the end of the main shaft, and operating a balanced poppet valve. The inertia governor, fig. 2,441, is used on the larger turbines. This type of governor affords a maximum amount of power and great sensitiveness. It is fitted throughout with ball bearings. The governor moves a pilot valve which directs steam to one or the other side of the piston in a steam cylinder, the piston rod moves a series of valves mounted on a single valve stem and operating in sequence. These valves admit steam to one or more sections of the nozzle at boiler pressure, throttling or wire drawing being eliminated.

P, to be opened in rapid succession. The result is that the piston N, is alternately raised and lowered, thereby opening and closing the admission valve. As the governor increases or decreases its speed, the movable fulcrum F, rises or falls and thus the relay valve M, is given a movement sufficient to raise the admission valve, and hold it open just the proper length of time. Since the valve M, is in mid-position when the full supply of steam is being admitted, any increase of speed at once shuts off the flow.

Reaction turbines are sometimes fitted with a by pass valve as shown in fig. 2,442, under direct control of the governor, and which opens for overloads above the range of the regular control nozzles, admitting steam at some intermediate point.

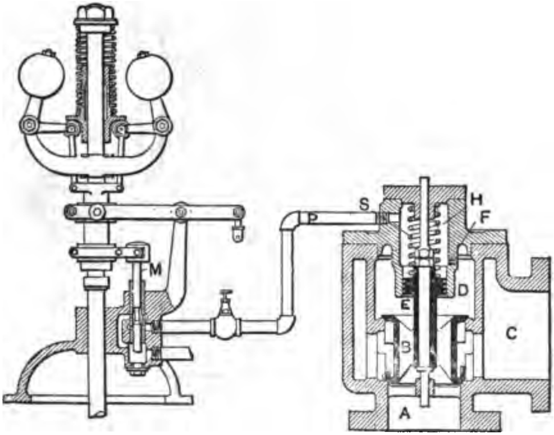


FIG. 2,442.—Westinghouse auxiliary governing by pass valve, admitting to the second drum for overloads. A, is the steam inlet, and B, valve which by passes steam to C. Steam can pass through B, to D, and act on the piston E. The equalizing passage F, causes the valve to remain closed by the spring S, under normal conditions. Connecting with the space H, above the piston E, is a pipe P, leading to a pilot valve M, under control of the governor. Under normal conditions the pilot valve remains closed. In case of overload, the pilot valve will open, allowing steam in H, to escape to the atmosphere, thus causing by pass B, to open and admit high pressure steam to the low pressure valve.

Working Pressures for Turbines.—To meet the varied conditions of service, turbines are designed to operate with:

1. High pressure;
2. Low pressure;
3. Mixed pressure.

Where all the power is furnished by the turbine, high pressure is used, but when operated in combination with a reciprocating engine, it is constructed to work with low pressure for constant load, or with mixed pressure for variable load.

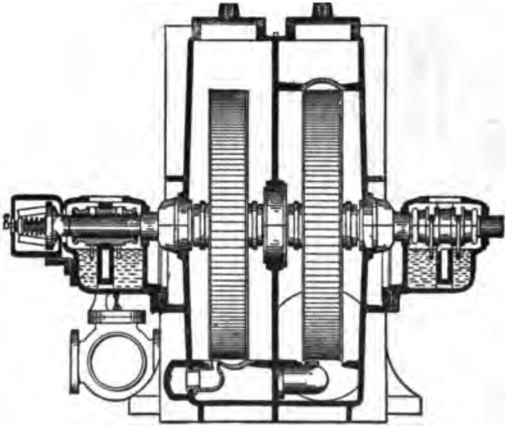
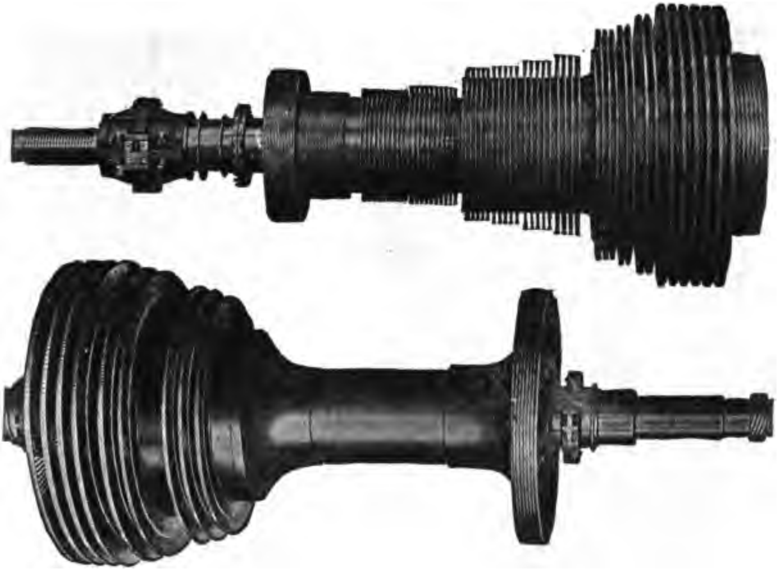


FIG. 2,443.—Terry compound two stage radial flow turbine for direct connected sets up to 300 kw. The vanes are semi-circular; steam escaping from the jet strikes the moving vanes, leaving the opposite side and is reversed by the stationary vanes to again enter the moving vanes at a point adjacent to the jet. This operation is repeated several times in the two stages. The governor is of the fly ball type, mounted directly on the turbine shaft, and controls a mitre throttle valve. According to builder's tests the water consumption (for brake horse power) is from 42 to 46 lbs. full load with superheated steam.

High pressure turbines operate at about the same initial pressures as triple expansion engines.

Low pressure, as here applied, means the terminal or exhaust pressure of the reciprocating engine, from which the exhaust steam passes through the turbine before entering the condenser. A low pressure turbine may be adapted to a variable load by the use of an *accumulator*.

Mixed pressure implies that the exhaust steam is supplemented, for heavy loads, by the admission of live steam which acts in separate nozzles so that the full pressure is used to create useful velocity.



FIGS. 2,444 and 2,445.—Allis-Chalmers, Parsons type, turbine rotors; figs. 2,444, 1,500 and 1,800 *r.p.m.* ***high pressure*** condensing turbine rotor; figs. 2,445, 3,600 *r.p.m.* ***low pressure*** condensing turbine rotor. ***In construction,*** in the large diameters, the rings which carry the blades are made separate from the body of the spindle as are the balance pistons. These parts are provided with tapered fits and are assembled by pressing. Each blade is formed by special tools, so that at its root it is of angular dovetail shape.

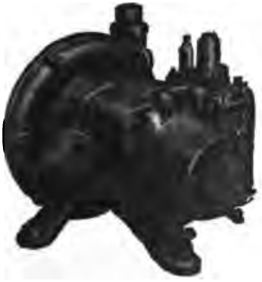


FIG. 2,446.—General Electric turbine driven dynamo for locomotive head light. ***It consists of*** a single stage Curtis turbine direct connected to a compound wound dynamo governed on the steam end by a self-contained pressure regulating valve, and controlled electrically by a stationary magnetic brake coil and rotating copper disc to maintain constant voltage on variable load.

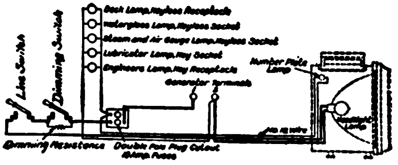

Line Switch
Dimming Switch
Deck Lamp, Keyless Receptacle
Waterglass Lamp, Keyless Socket
Steam and Air Gauge Lamp, Keyless Socket
Lubricator Lamp, Key Socket
Engineers Lamp, Key Receptacle
Number Plate Lamp
Generator Terminals
Headlight Lamp
No 12 Wire
Dimming Resistance
Double Pole Plug Cutout
10 Amp. Fuses

FIG. 2,447.—General Electric wiring diagram for locomotive head light. Usual pressure 6.5 and 33 volts.

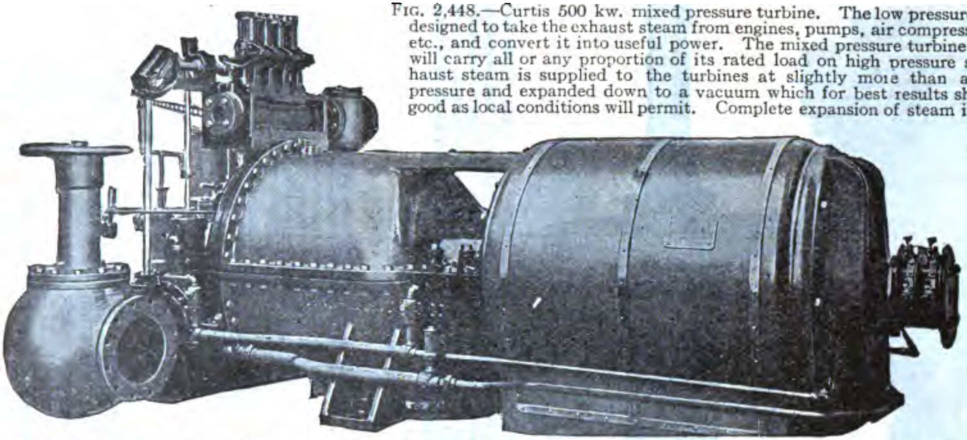


FIG. 2,448.—Curtis 500 kw. mixed pressure turbine. The low pressure turbine is designed to take the exhaust steam from engines, pumps, air compressors, hoists, etc., and convert it into useful power. The mixed pressure turbine in addition will carry all or any proportion of its rated load on high pressure steam. Exhaust steam is supplied to the turbines at slightly more than atmospheric pressure and expanded down to a vacuum which for best results should be as good as local conditions will permit. Complete expansion of steam in an engine is limited on account of cylinder condensation, wire drawing through the ports and practical sizes of low pressure cylinders, which render extra high vacuum of small value. On the other hand, the flow of steam through a turbine is continuous, and there is, consequently, no action corresponding to cylinder condensation, caused by steam striking a cold surface. Wire drawing is eliminated and sufficient space provided for the steam to expand to condenser pressure by increasing the size of buckets and nozzles. Low and mixed pressure turbines have been installed in power plants of all descriptions with substantial increase in economy. According to the builders, in the case of a plant operating non-condensing a low or mixed pressure turbine will develop from 80 per cent to 100 per cent as much power as the original engines without increasing the coal consumption. In the case of plants already operating condensing, the engines can in general be operated at full load non-condensing with an increased steam flow and the exhaust passed through a low or mixed pressure turbine will develop sufficient power to make a net gain of from 25 to 40 per cent. These results are possible because of the fact that a turbine is able to economically expand steam to a greater degree than a reciprocating engine as explained by the diagram, fig. 2,475. Curtis mixed pressure turbines may be divided into ***two main groups.*** ***First,*** those which are essentially low pressure turbines, designed for high economy when operating on exhaust steam, but provided with special nozzles in which the steam is economically expanded from boiler pressure and acts directly on the same bucket wheels as the low pressure steam. These nozzles are automatically brought into action in case the supply of low pressure steam is for any reason insufficient for the power required for the turbine. ***The second group*** comprises turbines designed for conditions where the supply of low pressure steam is so intermittent or insufficient as to render economy with boiler pressure steam of first importance. These turbines have a separate high pressure element which, in conjunction with the low pressure element, gives good economy when operating on boiler pressure steam. This element runs idle when the turbine is operating on low pressure steam, **but the turbine parts are so arranged that it runs in a vacuum instead of in steam** at atmospheric pressure. **That is, the high pressure economy is obtained at the least possible sacrifice of low pressure efficiency.**

Low Pressure Turbines.—The particular field which seems best suited to the turbine is its combination with the reciprocating engine, operating with the exhaust steam from the latter. The turbine, thus connected, makes available the energy of steam at pressures lower than the economical range of the steam engine. The reason for this is clearly explained by Prof. J. E. Denton as follows:

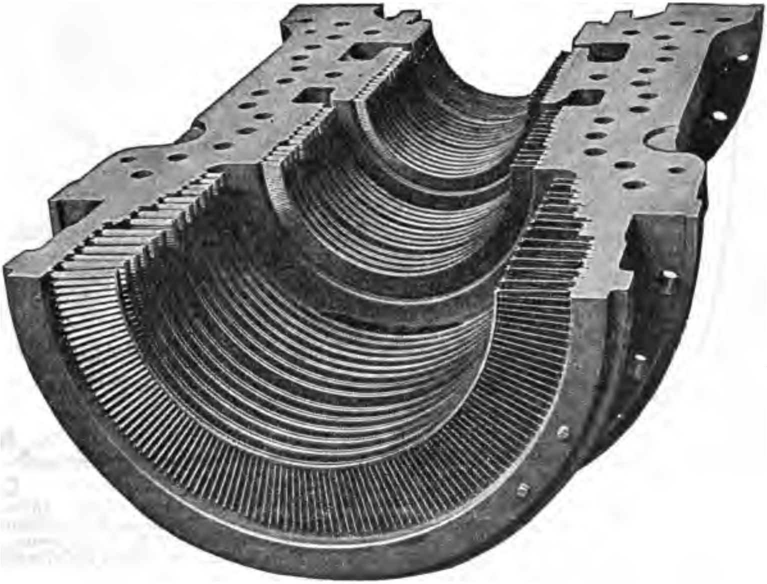


FIG. 2,449.—Half of Allis-Chalmers Parsons type, turbine cylinder showing blading installed.

"The most economical piston engines, which, as we have seen, expand steam about thirty times, release their steam at the end of the stroke at 6 pounds absolute pressure, but they exhaust into a condenser which, with a vacuum of 28″, contains a pressure of about one pound absolute per square inch. If the expansion could continue until the pressure of one pound was attained before exhaust occurred, considerably more work would be obtained from the steam. This, however, cannot be done in piston engines for two reasons: First, because the low pressure cylinder would have to be five times greater in volume, which is commercially impracticable; and second, because the velocity of exit through the largest exhaust ports possible is so great that the frictional resistance of the steam causes a pressure from one to three pounds higher than the condenser pressure in the best engines of ordinary piston speed.

"The steam turbine expands its steam to the pressure of the exhaust chamber, and as unlimited escape ports can be provided from this chamber to a condenser, it follows that the turbine can practically expand its steam to the pressure of the condenser."

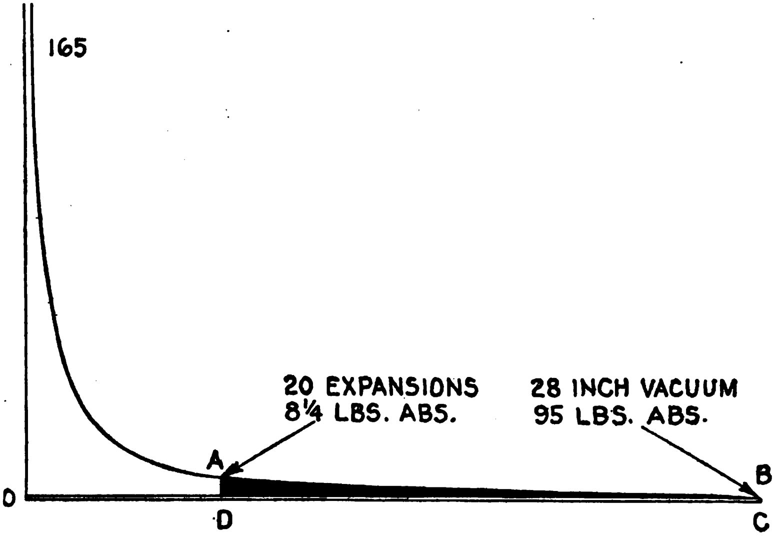


Fig. 2,450.—Card showing energy in steam not available for the reciprocating engine. In the turbine, steam is expanded to condenser pressure, whereas the terminal pressure in a steam engine must be several pounds higher, otherwise the low pressure cylinder would assume enormous proportions. This leaves a triangular area ABD, which represents useful work in the turbine, but which is lost in the steam engine.

The energy in steam not available for the reciprocating engine is shown by the black section of the expanding diagram, fig. 2,450. The curve shows the expansion of saturated steam from 150 lbs. gauge pressure to a vacuum of 28 inches. A compound condensing engine expanding steam twenty times would utilize the energy represented by the white area; now, if the exhaust steam be first passed through a low pressure turbine before entering the condenser the additional energy represented by the shaded section ABCD, which is rejected by the steam engine, is turned into useful work by the turbine.

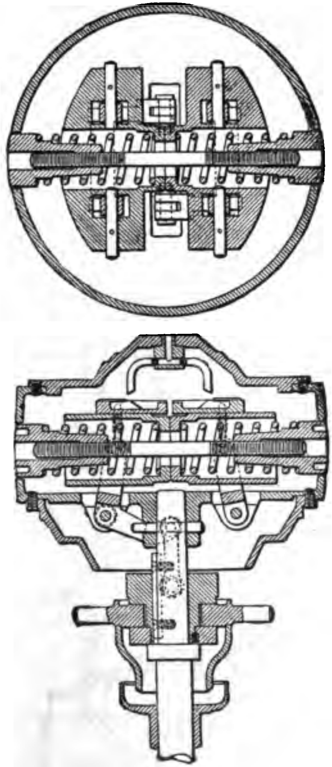
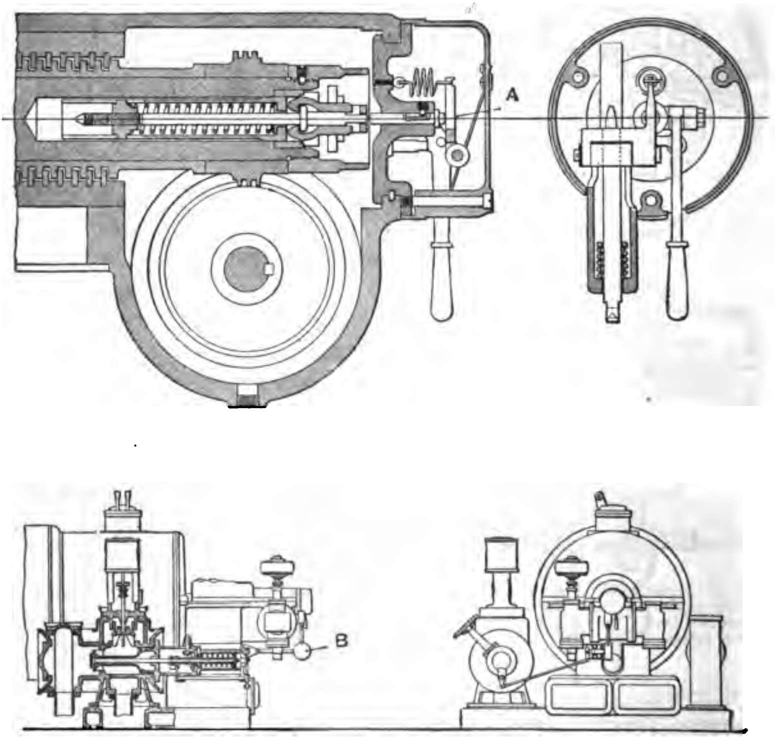


FIG. 2,451.—Allis-Chalmers, Parsons type, governor. The governor head as shown is of the spring type with equilibrated levers. It operates a balanced throttle valve by means of an oil controlled relay valve. The sensitiveness of the governor is made such as to secure the best results in the parallel operation of alternators. The governor can be adjusted for speed while the turbine is running, thus facilitating the synchronizing of alternators and dividing the load as may be desired.

It should be noted, from the diagram, that practically half the energy in the steam is spent in expanding to atmospheric pressure, and the other half in the remaining expansion. The advantage of using the turbine in combination with the reciprocating engine is clearly seen by comparing the black and shaded areas of the diagram, that is, without the turbine the black area is lost.

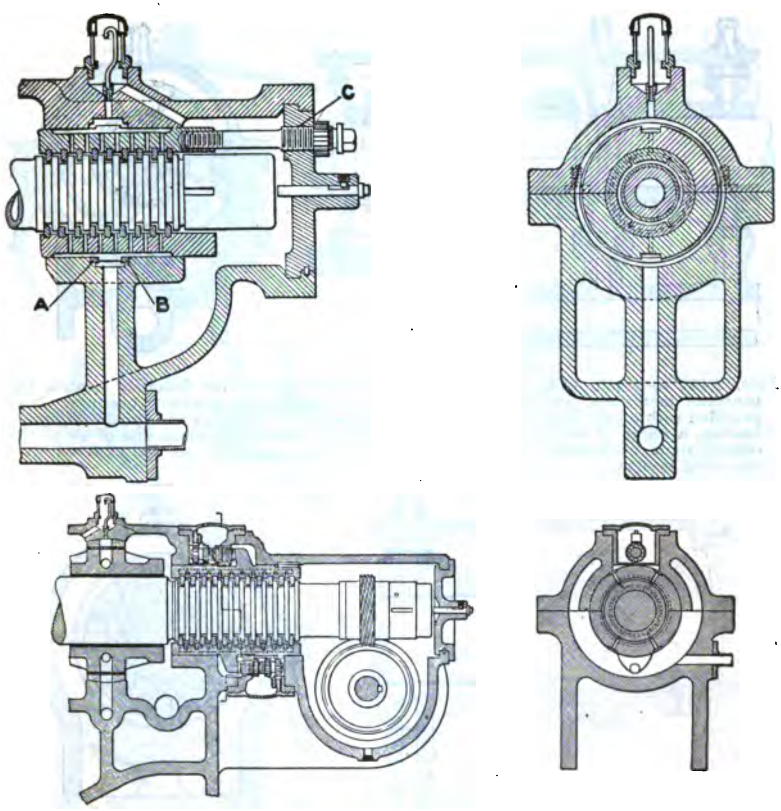
Rateau Accumulator.—There are numerous plants in which the supply of steam is intermittent, and of widely varying quality, such as, in rolling mills, for blooms, plate, sheet, wire, rail, and structural shapes. The majority of engines for such service operate under wasteful conditions.

Prof. Rateau nas devised an *accumulator* intended to regulate the intermittent flow of steam, storing up the excess, and giving it off during the non-flow periods, thus furnishing a uniform flow for the operation of a low pressure turbine. Prof. Rateau describes the working principle of his accumulator as follows:

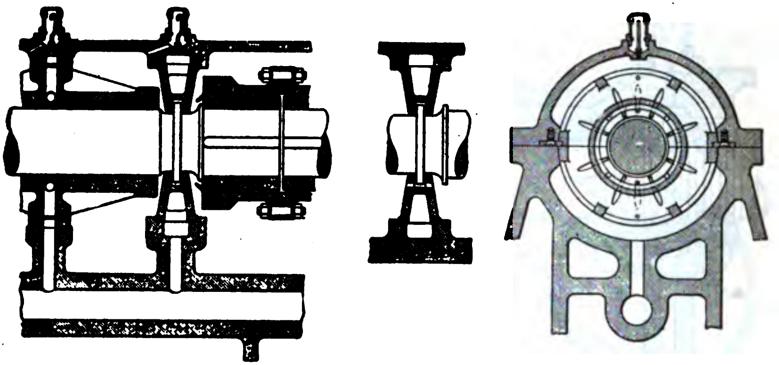


Figs. 2,452 and 2,453.—Allis-Chalmers Parsons type turbine safety stop and main throttle valve control mechanism. This over speed governor is located in the end of the turbine shaft, and is so arranged that when the predetermined over speed is reached, it will act, releasing the trigger A, and tripping the weight B, thus tripping main throttle valve, and shutting off the steam.

"The steam collects, and is condensed as it arrives in large quantities in the apparatus, and is again vaporized during the time when the exhaust



FIGS. 2,454 to 2,457.—Allis-Chalmers spindle adjusting bearing. The principal object of this collar bearing is to locate the turbine spindle accurately in an axial direction. For this reason it is made adjustable. This bearing also takes up such slight unbalanced thrust as may occur. These thrust bearings are made in two types. For the smaller machines, the lower half of the thrust bearing is locked in position by means of rings A B, fig. 2,454. The upper half bearing is adjusted by means of a differential screw C. In the larger sizes a slightly different arrangement is necessary as shown in figs. 2,456 and 2,459. In this case both upper and lower thrust blocks are made adjustable by means of screws, and, in order to prevent tampering with the setting, means are provided whereby a change of adjustment is made difficult, and the adjusting mechanism can be locked up if desired. Like the main bearings, these thrust bearings are automatically lubricated by a flood of oil, the oil being admitted between each thrust ring at two or more places, depending upon the size of the unit. As in the case of the bearings, a sight oil vent is provided upon the thrust bearing cover connecting to the oil feed system to prevent any accumulation of air at this point and to enable the operating engineer to see at any time that the thrust bearing is receiving its full share of lubricating oil.



FIGS. 2,458 to 2,460.—Allis-Chalmers generator thrust bearing for definitely locating the generator motor and preventing same floating up against the generator bearings. It is provided with ample lubrication, and, as in the case of the bearings and spindle adjusting bearing, a slight oil vent is provided on the cover to prevent accumulation of air in the oiling system at this point; this also enables the operating engineer to see that all is well with the oiling system.

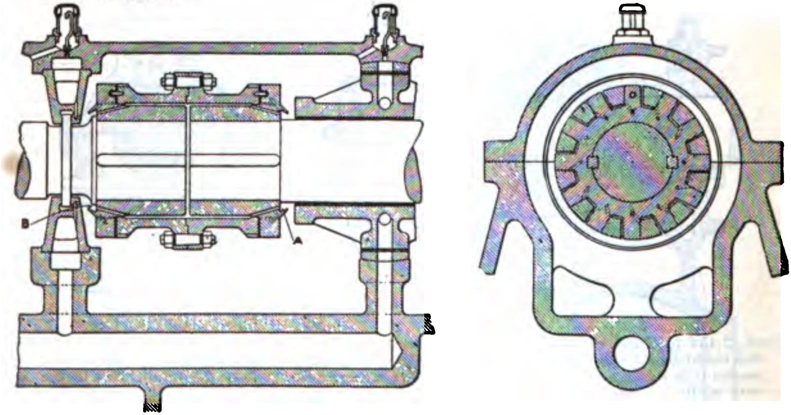
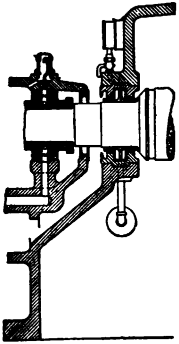
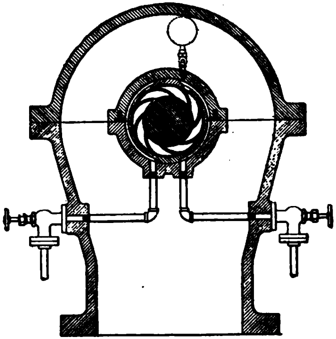
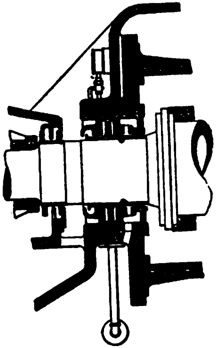
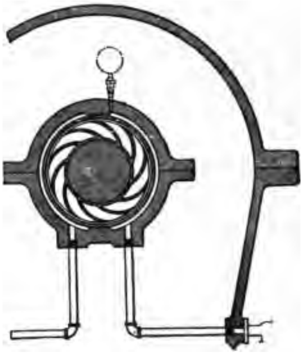


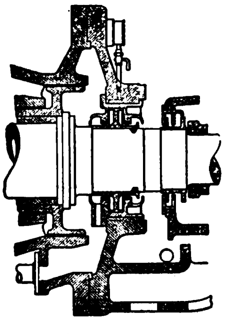
FIG. 2,461 and 2,462.—Allis-Chalmers flexible coupling. This is used to provide for any slight inequality in the wear of the bearings, to permit axial adjustment of the turbine spindle, and to allow for differences in expansion. This coupling is so made that it can be readily disconnected for the removal of the turbine spindle or of the revolving field of the generator. Provision is made for ample lubrication of the adjoining faces of the coupling. This is effected at the turbine end of the coupling by means of oil escaping from the turbine bearing flowing into the oil cup A, and from there finding its way to the contact faces through holes provided for the purpose. At the generator end oil is furnished to oil cup B, from pressure oiling system and so to contact faces. The coupling is enclosed in the bearing housing so that it is completely protected against damage and cannot cause injury to the attendants.

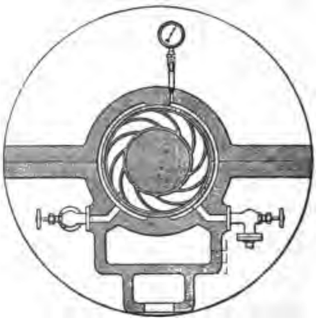








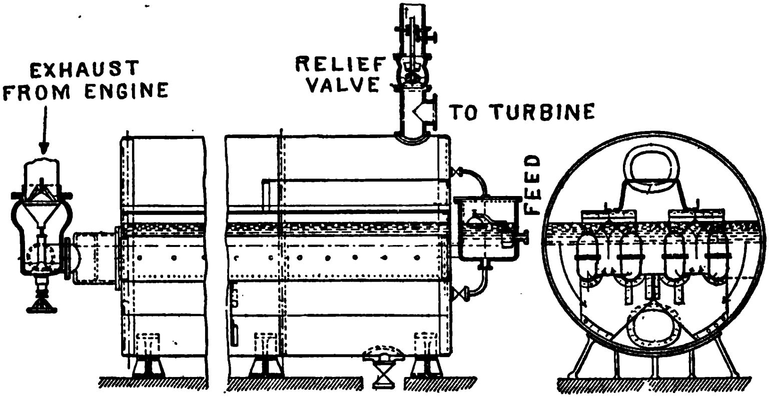




FIGS. 2,463 to 2,468.—Allis-Chalmers spindle glands. Where the turbine shaft passes out of the cylinder it is necessary to provide against the inleakage of air or outleakage of steam. To accomplish this, a water gland is used. ***It consists of*** a paddle wheel fixed on the turbine spindle and revolving in a specially constructed casing, figs. 2,463 and 2,464. Water is introduced in this casing and the high number of revolutions causes a perfect seal with comparatively little friction. In the larger types of turbine where the low pressure piston is located at the exhaust end of the turbine, a slightly different type of gland is necessary at the low pressure end owing to the difference in pressure at the two ends of the cylinder. The pressure at the gland at the *h. p.* end is that of the exhaust, while at the *l. p.* end it is that of the intermediate stage. As it is necessary for balancing purposes that the pressures on both ends be the same, a series of rings having small radial clearances are furnished in the *l. p.* gland casing forming a labyrinth packing between the intermediate and low pressure steam spaces, figs. 2,465 and 2,466. Figs. 2,467 and 2,468 show the corresponding ***h. p.*** **end gland casing. It is essential that the water for feeding these glands be free from acid and scale forming material and, in case such a supply is not available, a water purification system should be installed.**

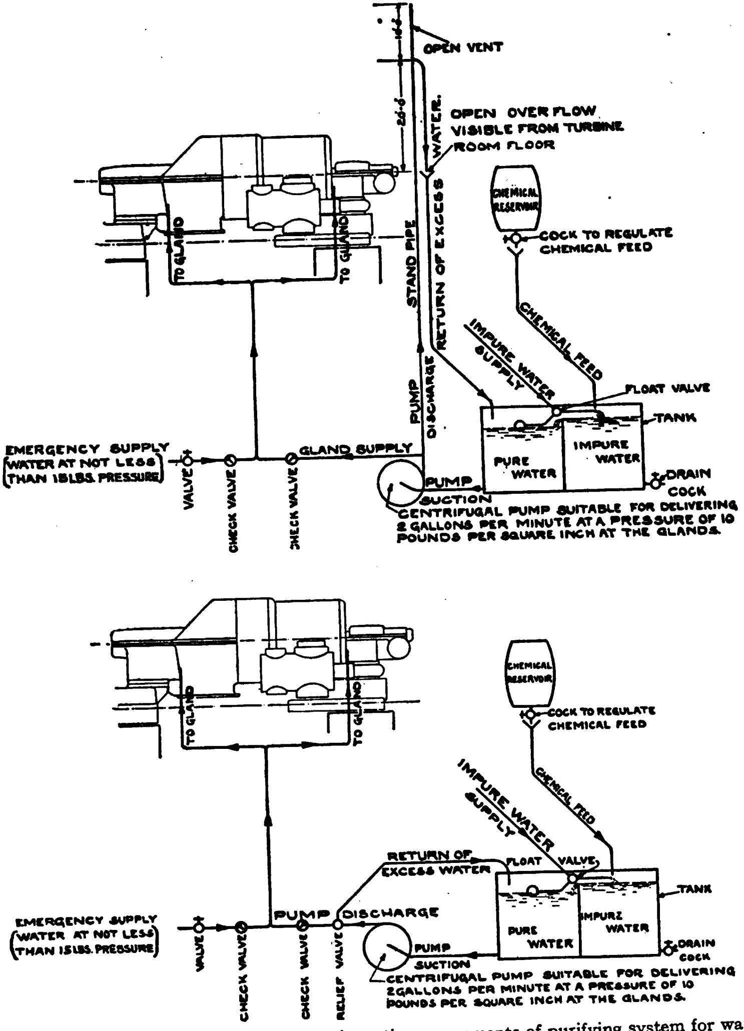
of the engine diminishes or increases. The necessary variations for condensation and regeneration of the steam correspond to the fluctuations in pressure in the accumulator, the pressure rising when the apparatus is being filled, and descending when it is discharging into the turbine. Water which has a very high heat capacity has been used as a fly wheel, but in order to rapidly communicate to a liquid mass a considerable quantity of heat, corresponding to the latent heat of steam to be condensed it becomes necessary, owing to the poor conductivity of the water, either to arrange it in thin layers, or to cause a rapid circulation in order to increase the surface of contact between the steam and the water itself."

In the first method, numerous flat cast iron trays were used, forming receptacles for the water. The second solution gave rise to an accumulator consisting of a cylindrical vessel partly filled with water, and having a perforated exhaust pipe below the water line, as shown in fig. 2,469.

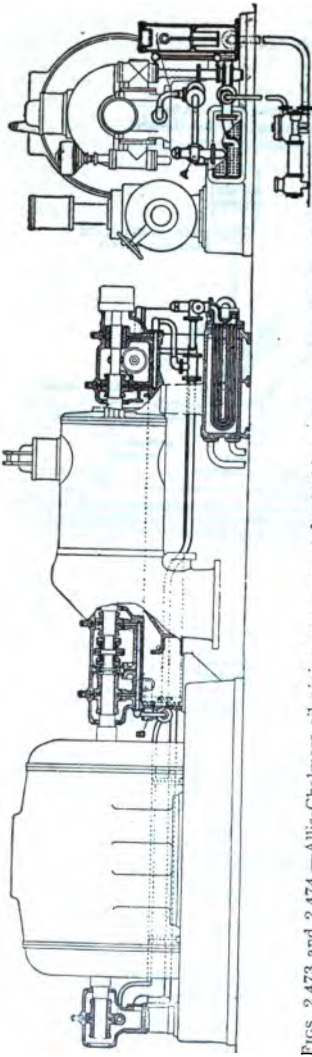


FIGS. 2,469 and 2,470.—Rateau accumulator for low pressure turbines. This device is used where the steam supply for the turbine is intermittent, as in rolling mills. Any excess heat is stored up in heating the water, to be later given off when the supply fluctuates below that required for the turbine.

Accumulator Accessories.—For the successful operation of accumulators, they should be fitted with several accessories as follows:


OPEN VENT
16-6
26-6
OPEN OVER FLOW VISIBLE FROM TURBINE ROOM FLOOR
RETURN OF EXCESS WATER
STAND PIPE
PUMP DISCHARGE
TO GLAND
TO GLAND
CHEMICAL RESERVOIR
COCK TO REGULATE CHEMICAL FEED
CHEMICAL FEED
IMPURE WATER SUPPLY
FLOAT VALVE
TANK
PURE WATER
IMPURE WATER
DRAIN COCK
EMERGENCY SUPPLY (WATER AT NOT LESS THAN 15 LBS. PRESSURE)
GLAND SUPPLY
VALVE
CHECK VALVE
CHECK VALVE
PUMP
SUCTION
CENTRIFUGAL PUMP SUITABLE FOR DELIVERING 2 GALLONS PER MINUTE AT A PRESSURE OF 10 POUNDS PER SQUARE INCH AT THE GLANDS.
TO GLAND
TO GLAND
CHEMICAL RESERVOIR
COCK TO REGULATE CHEMICAL FEED
CHEMICAL FEED
IMPURE WATER SUPPLY
RETURN OF EXCESS WATER
FLOAT VALVE
TANK
PURE WATER
IMPURE WATER
DRAIN COCK
EMERGENCY SUPPLY (WATER AT NOT LESS THAN 15 LBS. PRESSURE)
PUMP DISCHARGE
VALVE
CHECK VALVE
CHECK VALVE
RELIEF VALVE
PUMP
SUCTION
CENTRIFUGAL PUMP SUITABLE FOR DELIVERING 2 GALLONS PER MINUTE AT A PRESSURE OF 10 POUNDS PER SQUARE INCH AT THE GLANDS.

FIGS. 2,471 and 2,472.—Allis-Chalmers schematic arrangements of purifying system for water for glands of steam turbine. Fig. 2,471, first arrangement; fig. 2,472, second arrangement.



FIGS. 2,473 and 2,474.—Allis-Chalmers oil piping arrangement for turbine. The oil tank is located at the back of the turbine, the oil strainer being bolted to one end. The oil pump operated from the turbine shaft by means of cut gears working in an oil bath, is located at the thrust end of the turbine below the level of the oil tank and receives the oil from the oil tank after it has passed through the oil strainer, delivering it under a pressure of from 25 pounds to 30 pounds to the oil cooler, where it is sufficiently reduced in temperature before passing on to the bearings, etc. After leaving the cooler, part of the oil is used for the oil relay system of governing the turbine, and the remainder, reduced in pressure to about 3 pounds or 4 pounds by the reducing valve A, is delivered to the various bearings. Any excess in pressure in the oiling system is taken care of by relief valve B, which by passes the oil back to the oil tank. An auxiliary oil pump is furnished for use when starting up or stopping the turbine, or for use in case of emergency. On the smaller machines this auxiliary oil pump is placed on the generator bed-plate at the back of the turbine. On the larger turbines the location of this pump is left to the purchaser, who may place it in any convenient position close to the turbine.

1. *Automatic relief valve* to by pass the steam from the engine to the condenser when the turbine does not require all of the steam exhausted by the engine.

2. *Automatic expansion valve* to admit live steam from the boiler to the turbine, should sufficient exhaust steam be not available, or in case of temporary shut down of the main engine.

3. *Steam check valve* to shut off the accumulator from the turbine when the engine is shut down and the turbine is supplied only with live steam.

4. *Water check valve* to prevent water in the accumulator returning toward the engine through the exhaust supply pipe when the engine is shut down.

5. *Drain valve* to carry off any excess water that may collect.

NOTE.—***Steaming of turbines.*** In heating up turbines for steaming purposes, if the turbines be of ***large size,*** it is usual to have heating steam pipes tapped into the casing at the turbine end. In ***small*** or ***average size,*** the steam is admitted at one end and passed out at the other. Some prefer to heat up the turbine as quickly as possible to ensure equal expansion all over, and others, to heat up gradually. Steam being admitted into casing, the rotor should be turned about every two minutes, to avoid heating up one side of the rotor excessively, which would cause it to expand on that side, thus disturbing the alignment of the shaft and cause it to bend in the bearings.

Importance of High Vacuum.—The economy due to working steam with very low back pressures is more marked in the case of the turbine than with the steam engine. The reason for this is illustrated in fig. 2,475.

The diagram shows the expansion of one pound of steam from 100 lbs. pressure down to one pound absolute (27.9 vacuum) and also the work done both before and during the expansion, as in an indicator card.

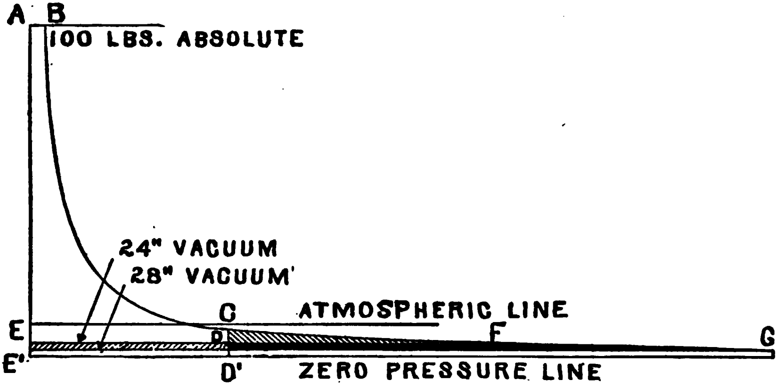


FIG. 2,475.—Card showing the importance of high vacuum for economical operation of a turbine. A steam engine would release steam at some point C considerably above the exhaust line, but in the turbine the expansion is continued to condenser pressure. If the vacuum be increased say, from 24 inches to 28 inches, the gain in power for the steam engine would be represented by the shaded area E D D′ E′, and for the turbine by the solid black area D F G D′. The comparative sizes of these areas show forcibly the importance of high vacuum for the turbine.

In a condensing engine, owing to the limited size of the cylinder, the expansion of the steam would be interrupted at some point C, a little below the atmospheric line, by the opening of the exhaust valve, and the pressure would suddenly fall to that of the condenser. With a 24″ vacuum the work done is represented by the shaded card A B C D E.

If a turbine be interposed between the engine and the condenser, expansion will continue from C to F, expanding down to the condenser pressure. The gain in power due to the turbine then, is represented by the shaded area, C F D.

Now, if the vacuum be increased from 24″ to 28″, the gain in power for the engine would be that due simply to the reduction in back pressure indicated by the shaded area E D D′ E; similarly for the turbine the solid black area D F G D′ will represent the increase in power. It should be noted that in the case of the turbine the gain is considerably more than for the engine, hence the importance of operating the former with very low back pressure. The gain in power due to the lower back pressure is

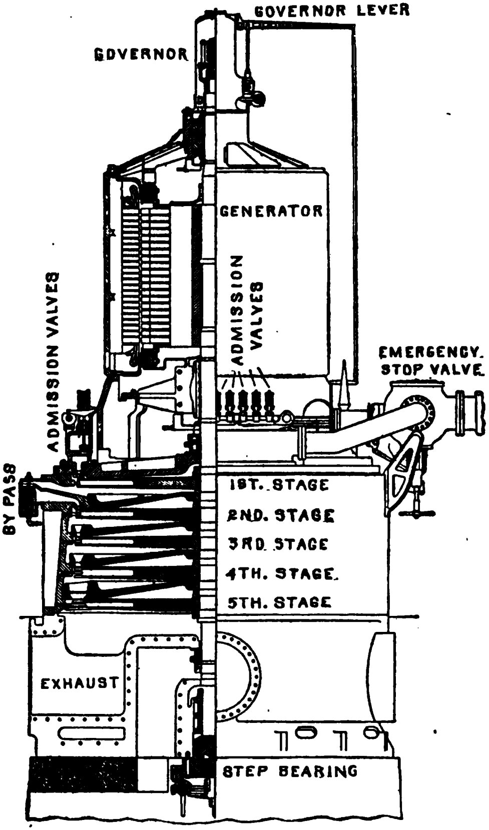


FIG. 2,476.—Curtis vertical compound multi-stage impulse turbine. As shown, there are five stages, each stage is compounded by having two sets of moving vanes separated by a set of stationary vanes. Superposed on the turbine is the generator, and on top, the governor which controls the speed by operating the admission valves indirectly with electro-magnets. It governs by partial admission. A by pass or stage valve (shown at left) automatically by passes steam from the first to the second stage for overload.

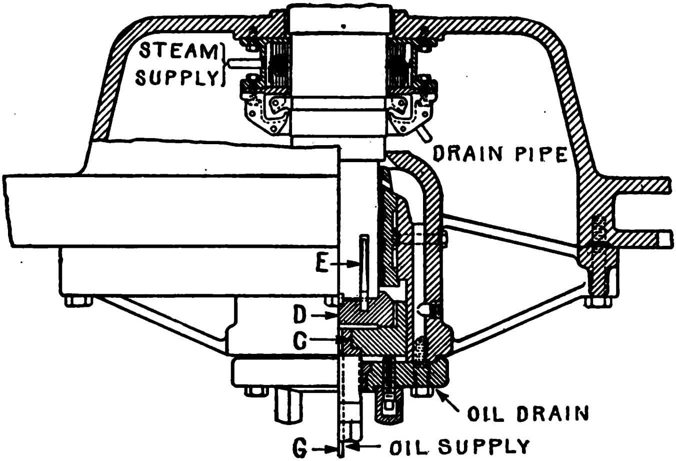


FIG. 2,477.—Curtis step bearing. There are two cast iron blocks, C and D, one carried by the end of the shaft, and the other being adjustable by set screws. Both blocks are recessed, forming a space into which oil is forced through a central bore, with sufficient pressure to raise the shaft and support its weight on a thin film of oil. The lubricant oozes out through the bearing to the overflow.

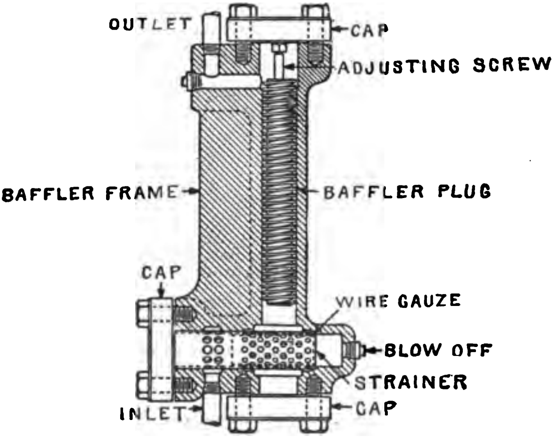


FIG. 2,478.—Curtis step bearing baffler for regulating the pressure at the step bearing. It consists of a long screw plug inserted into the body of the device and past which the oil must flow to the step bearing. The more turns, the greater the resistance offered to the flow.

partly offset by the extra cost of maintaining the higher vacuum, and with the engine by the additional loss due to increased condensation.

The increase in vacuum shown in the diagram necessitates a large increase in the size of condensing apparatus required. The proportions of the cooling surface, and circulating water must be about doubled, the air pump enlarged, etc., all of which means a higher first cost, increased operating expenses, and an additional sum to cover the interest on the larger investment. In practice, these items limit the degree of vacuum to about 28½″ to 29″ for the turbine and 27″ to 28″ for the engine.

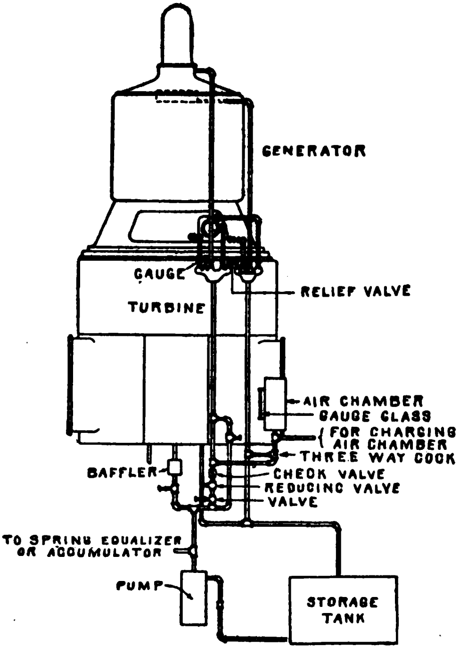
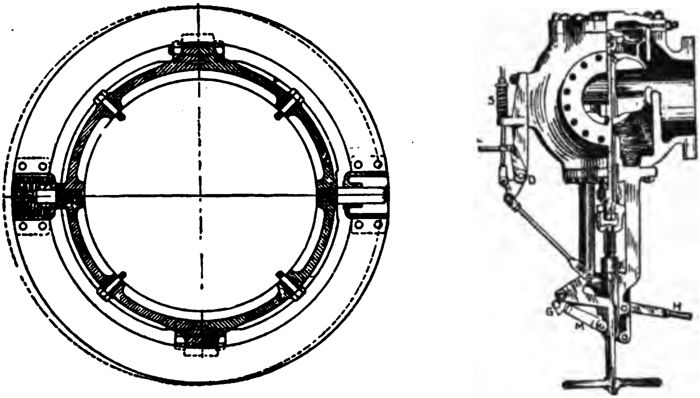


FIG. 2,479.—Curtis forced lubrication system. A tank of sufficient capacity to contain all the oil is fitted with suitable straining devices and a cooling coil, and located at a level such that the oil will return by gravity from the various points. A pump maintains a pressure about 25 per cent higher than that required for the step bearing. The step bearing baffler regulates the pressure for the step bearing, and a reducing valve for the other parts of the system, which reduces the pressure to about 60 lbs. The lubrication system, which includes a storage tank partly filled with compressed air, operates the hydraulic governor mechanism, and supplies oil to the upper bearings. A relief valve prevents excess of pressure. Visible drains lead from the hydraulic cylinder, upper bearings, and relief valve, discharging into a common chamber.

Starting a Curtis Turbine.—Although a turbine has no eccentric to slip or cylinder head to knock out, the same care should be exercised as in starting a steam engine, for there are hundreds of small buckets which run at a speed of nearly five miles a minute, and they are liable to rub against the stationary blades if not properly adjusted.

1. Before starting, the step bearing pump must be put in operation so that the weight of the rotor may be supported by oil under pressure, thus separating the step bearing blocks.

Failure to start this pump may result in heating the step bearing blocks red hot, and cause damage.



Figs. 2,480 and 2,481.—Emergency governor and emergency stop valve. The governor consists of a ring placed in slightly eccentric position around the shaft between the turbine and generator. Any undue increase of speed causes an excess of centrifugal force, which overcomes the resistance of the spring and releases rod L, fig. 2,481. This rod is connected to a crank D; when released, the tension of the spring S, pulls up the gear and throws out the hook G, allowing the valve to close by gravity and unbalanced steam pressure.

2. Next, the circulating pumps should be started.

3. The auxiliary generator is now started, so that current may be available for field excitation and for operating the valves of the turbine.

4. The turbine is set in motion by throwing in the starting switch and slightly opening the throttle.

The starting switch puts in service three magnets which cause three valves to open; this admits sufficient steam to bring the machine up to speed.

5. As soon as the turbine is in motion the atmospheric valve and air pump should be started, also the hot well pump.

As the vacuum increases, the turbine will increase in speed more rapidly, and by opening the throttle until boiler pressure is obtained, it will come up to speed in a few minutes.

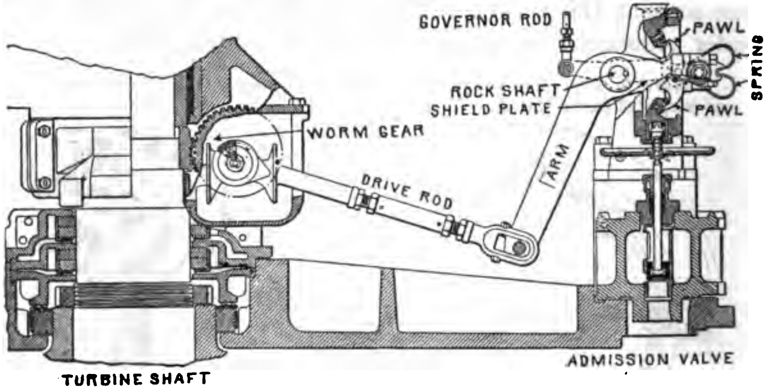


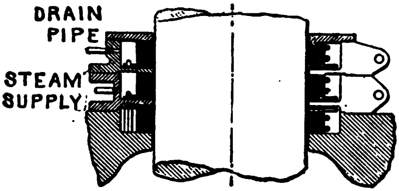
FIG. 2,482.—Curtis mechanical valve gear. The admission valves are located in steam chests directly over the first stage nozzles. The valve stems extend upward through stuffing boxes and are attached to notched cross heads. Each cross head is operated by a pair of reciprocating pawls, or dogs, one for opening, one for closing the valve. The several pairs of pawls are hung on a common shaft, which receives a rocking motion from a crank, driven from a worm and worm wheel by the turbine shaft. The pawls engage in notches in the side of the cross heads to open or close the valves, the engagement being effected by *shield plates* under control of the governor. These plates are set one a little ahead of the other, to obtain successive opening or closing of the valves. When more steam is required the shield plate allows the proper pawl to fall into its notch in the cross head and lift the valve from its seat. If less steam be wanted, the shield plate rises and allows the lower pawl to close the valve on the down stroke. The valve is held either open or shut by the steam pressure till it is moved by the pawl on the rock shaft. The amount of travel on the rock shaft is fixed by the design, but the proportionate travel above and below the horizontal is controlled by the length of the connecting rods from the crank to the rock shaft.

The hot well pump not only takes care of the condensation but the step bearing water also, which, after passing between the step, runs down through the condenser into the hot well.

6. When the governor begins to act the running switch should be closed.

This switch operates all the valves except those controlled by the starting switch.

As the governor begins to act it lifts the fingers off the roller, thereby



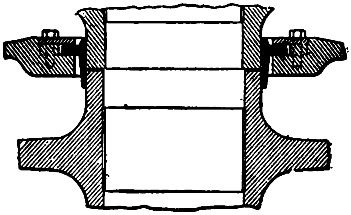


FIG. 2,483.—Curtis upper stuffing box. The packing consists of carbon rings of three segments each. One ring of segments breaks joints with its mate in the case; the rings are separated by flanges. In some cases the packing is kept from turning by means of a link, one end being fastened to the case and the other to the packing holder. The packing is held against the shaft by light springs, or steam pressure as shown. The space around the two upper sets of rings is drained to a third stage by means of a thread way cock, which keep the balance between the atmosphere and stuffing box pressure.

FIG. 2,484.—Curtis self-centering packing ring. This forms the joint at the center where the wheel hubs pass through from stage to stage. The packing is a close running fit, and is provided with grooves which break up and diminish the leakage of steam.

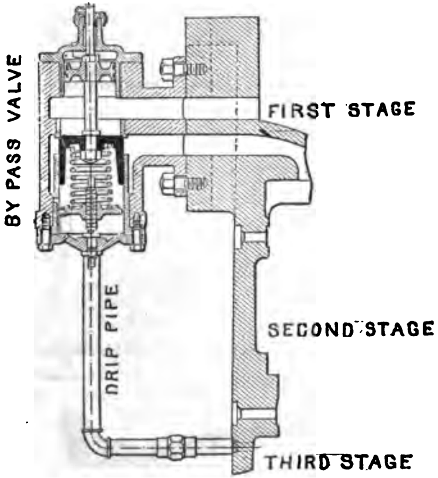
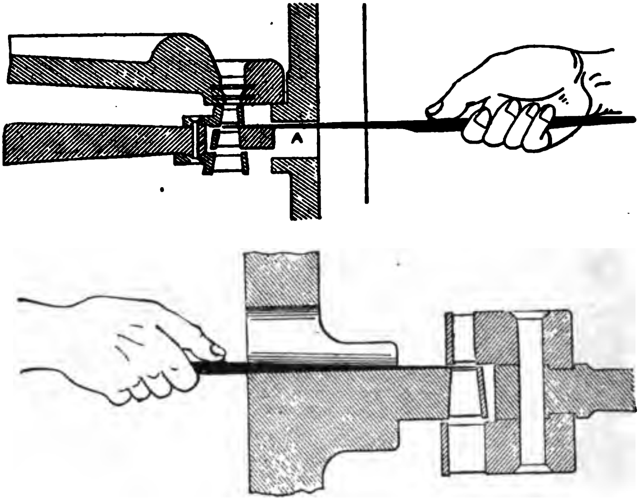


FIG. 2,485.—Curtis by pass or stage valve. This is automatic in its action and is used to prevent the pressure becoming too high in the first stage, in case of a sudden overload. It transfers a part of the steam to a special set of expansion nozzles over the second wheel, thus relieving the pressure in the first stage and using the steam in a lower stage to increase the power of the machine. Any leakage past the valve is taken care of by a drip pipe to the third stage.



Figs. 2,486 and 2,487.—Method of determining clearance in Curtis turbines. After removing the peep hole plugs, a taper gauge is inserted both above and below the moving vanes, and the distance it enters registers the clearance for each side. The peep holes are sometimes opposite the stationary vanes as in fig. 2,486. and sometimes opposite the moving vanes, as in fig. 2,487. ***It is imperative that there be no rubbing contact between the vanes while the machine is running.***

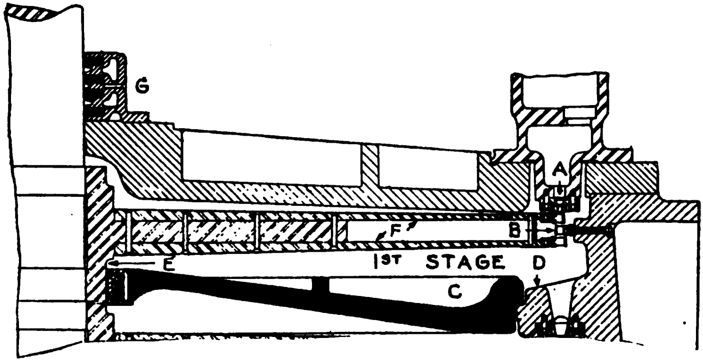
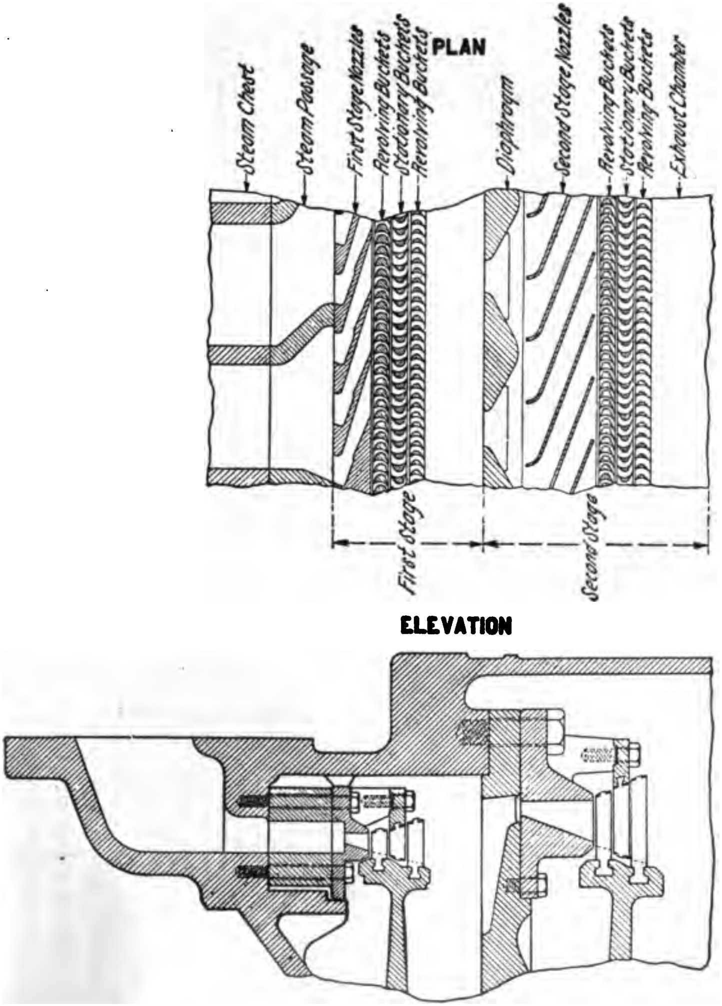


Fig. 2,488.—Sectional view, showing the construction details of one stage of a Curtis turbine. The parts are: A, admission nozzles; B, stationary vanes; C, diaphragm separating first and second stages; D, second stage nozzles; E, wheel hub; F, discs carrying moving vanes; G, upper stuffing box.



Figs. 2,489 and 2,490.—Plan and elevation, showing steam passages in a two stage Curtis steam turbine.

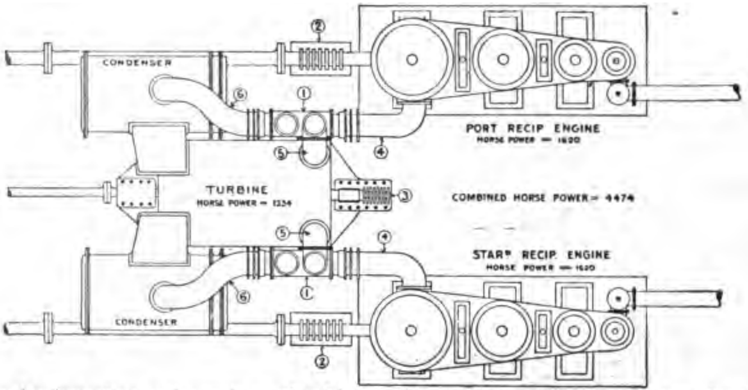
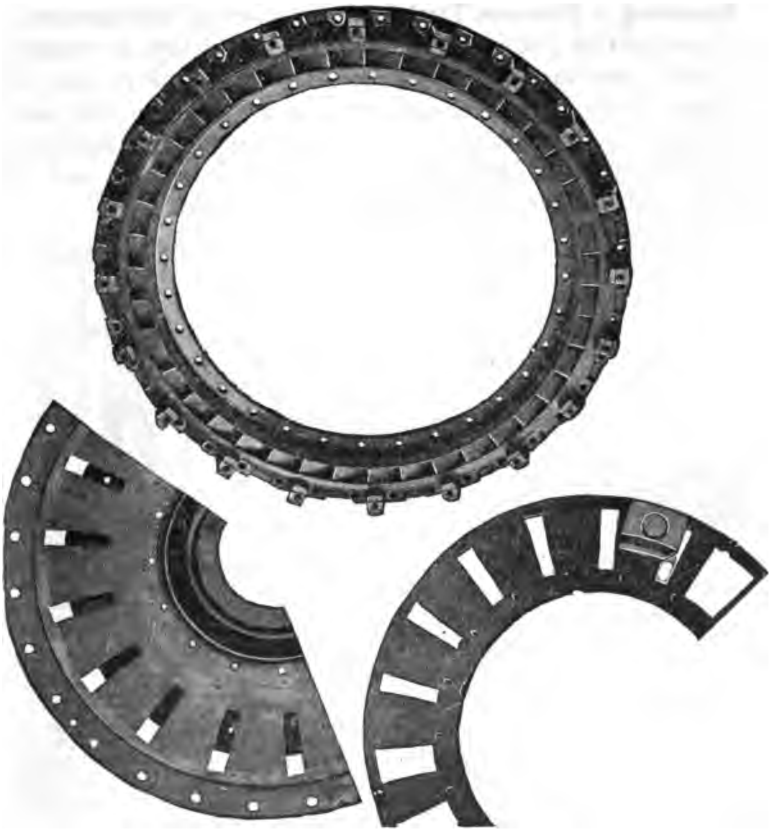


FIG. 2,491.—Combined reciprocating marine engines and turbines. ***The arrangement consists of*** two wing triple or quadruple expansion engines exhausting into a central low pressure ahead turbine, driving a third shaft and propeller, the revolution speed of the center turbine shaft being much higher than that of the wing shafts. An alternative design is that of two wing low pressure turbines and one center reciprocating engine. The engines are arranged to be run as follows: **1,** boiler steam to both *h.p.* cylinders, and exhaust from these to turbine, the exhaust from the turbine being divided and lead into two separate condensers; **2,** boiler steam to both *h.p.* cylinders, and exhaust from these to condensers direct, the turbine being then cut off. This is required when running astern as the turbine is for ahead running only, and may be used for ahead running with two propellers only; **3,** boiler steam to either *h.p.* cylinder, and exhaust from *l.p.* to center turbine, then into one condenser only. ***These combinations*** are obtained by the use of "change valves" fitted on the *l.p.* exhaust pipes, and by large butterfly valves fitted in the turbine exhaust branches. The change valves admit the reciprocating exhaust steam of either side to the turbine, or to the condenser as required, and the large valves in the turbine exhaust pipe shut off the condenser on either side as may become necessary should one reciprocating engine require to be disconnected through breakdown. ***The parts are:*** 1, change valve, giving steam either to turbine or condenser direct; 2, reciprocating engine thrust block; 3, turbine thrust black; 4, exhaust from reciprocating engine to turbine or to condenser; 5, branch to turbine; 6, branch to condenser.

opening the circuits to the magnets. When up to speed all of the fingers will have been lifted except one or two, or just enough to maintain uniform speed, depending upon the load.

The turbine, when running, requires the same careful attention as a steam engine, owing to the high speed of operation. Careful inspection should be frequently made for loose screws and nuts; any looseness in counterweights will throw the machine out of balance.



Figs. 2,492 to 2,494.—Curtis horizontal turbine construction details. Figs. 2,492, high pressure side of diaphragm; fig. 2,493, second stage nozzle; fig. 2,495, high pressure side ***extraction valve,*** for extracting steam for heating and manufacturing purposes. In the smaller sizes an opening is provided in the first stage from which moderate amounts of steam may be extracted when the generator is fairly well loaded. The amount of steam extracted is regulated by a valve placed in the extraction line. The larger sizes may be provided with an internal valve for controlling the admission of the steam to the later stages of the turbine, and thereby diverting a greater or less amount of steam to the extraction line. The internal valve is controlled either by means of a hand wheel, or by an automatic device. The steam which is extracted has passed through one or two stages of the turbine, and in so doing has given up some of its energy for driving the generator. The energy given up in this way is made available by expansion from boiler pressure down to the pressure at which it is extracted.

Running a Parsons Turbine.—In starting, the circulating pump should be put in operation while the turbine is warming up; next, the turbine should be started gradually, also the auxiliary generator. When the speed increases to the point where the load begins to come on, the gland water should be turned on, the hot well and dry air pumps started, and the turbine and auxiliary generator brought to speed.

If the turbine be brought to speed too quickly there will be considerable vibration.

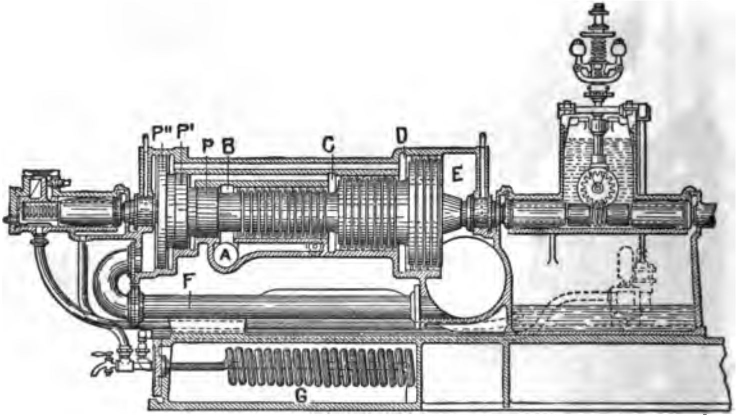


FIG. 2,495.—Westinghouse-Parsons turbine. Steam enters the chamber B, at boiler pressure through pipe A, and passes to the right through the first group of blades which gradually increase in height chamber C. Here, to avoid excessively long blades as well as many sizes of blades, it is necessary to jump to a larger diameter, and the steam flows through a second set to D, and finally through a third set to space E. The balancing pistons P P' and P", are of such diameter that the steam pressure against them balances the axial thrust in the direction of the steam flow. The diameters of the pistons are approximately equal to the mean diameters of the steam areas of the different steps. The pipe F, connects the space back of the balancing pistons with the exhaust chamber. G, is a coil for cooling the oil circulating through the bearings.

If a Westinghouse-Parsons turbine be started with a vacuum, leakage of cold air will occur through the glands, tending to set up unequal temperature conditions in the turbine, because the glands, which are water packed, do not become sealed until the machine is in operation. If, in starting, steam should leak through the glands, water should be turned on for a second or two, and then shut off. This supply will prevent steam leaking through until the turbine is up to speed.

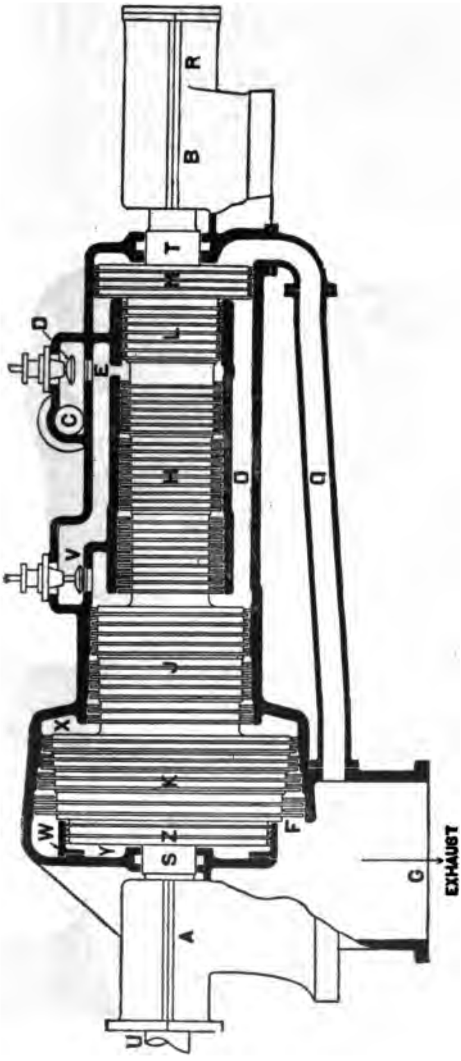
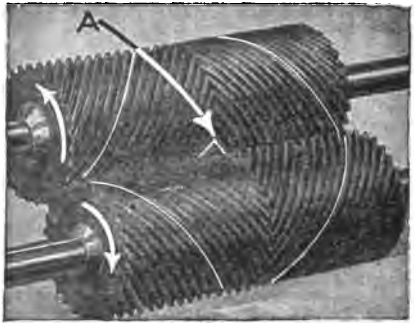
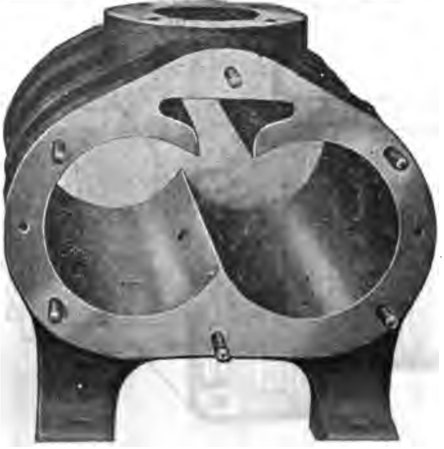


FIG. 2,496.—Allis-Chalmers-Parsons turbine. Steam enters at C, and is admitted through the main throttle D; it passes through passage E, and turns to the left, acting on the numerous rows of blades and emerges at G, to the condenser. The leftward thrust of the steam is counteracted by balance pistons L, M, Z, the latter working inside a supplementary cylinder W. The pressure on the balance piston at Y, is equalized with that of the third series of vanes at X, by passages (not shown) through the body of the spindle. S and T, are water packed glands to prevent leakage of steam. For overload, a by pass valve V, is used, which admits high pressure steam to the second series of vanes.

In shutting down, the gland water should be shut off and the drips opened, as soon as the vacuum drops. When the turbine is in operation, attention should be given chiefly to the oil supply and the gland water.

The weight of the rotor is supported by the bearings, causing considerable heating. The high temperature makes necessary a cooling coil for the oil, as shown in fig. 2,495; this must be cleaned as often as required to keep the coil surfaces effective. The variable temperature causes some oils to deteriorate, thus rendering them unfit for lubrication. In some cases the oil, when it becomes





FIGS. 2,497 to 2,500.—Buffalo "Spiro" turbine. ***It consists of*** two herringbone gear wheels meshing together and revolving in a close fitting casing. The two gears or rotors are shown in fig. 2,497, and their casing in fig. 2,498. It is notable that the herringbone rotors are much longer axially in proportion to their diameter than ordinary herringbone gears used for power transmission. Steam is admitted at mid-length into the tooth pockets at the point A, of each rotor. Thus the steam occupies the space between two adjacent teeth, and this space is closed at the tooth points by the closely fitting casing. As each rotor turns, the tooth space occupied by the steam increases in length and the steam expands. Finally the steam escapes when the outer ends of the teeth pass the line of contact between the two rotors. The increase in length of the tooth space from the time when steam is first admitted until the exhaust occurs, is illustrated by the length of the outer white lines in fig. 2,497, as compared with the length of the grooves at point A. The usual arrangement, as shown by fig. 2,498, is to have the steam inlet on the under side of the rotors, and the exhaust on top. In this way the weight of the rotors is partly carried by the steam pressure, which is greatest on the under side, decreasing towards the exhaust on top. The inlet port openings, as indicated in fig. 2,498, are situated one on each side of the central rib, causing the fluid to impinge against the teeth of both rotors simultaneously. The central rib divides the casing into two partly closed cylinders. The ends of these cylinders are covered by the heads, which are shown separately in figs. 2,499 and 2,500. The heads contain the bearings and packing glands. Drive can be taken from any or all of the shaft ends.

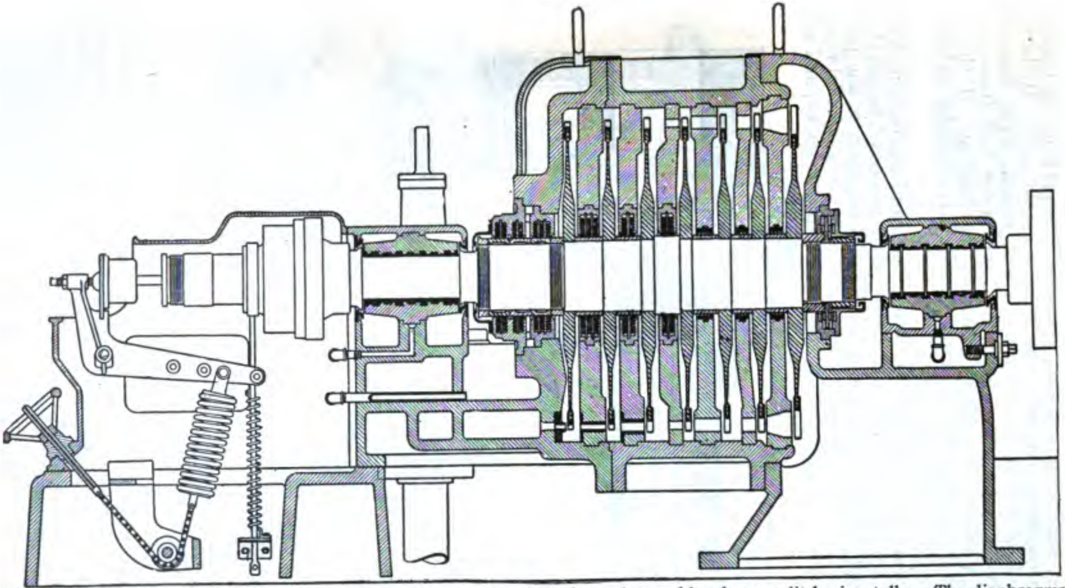
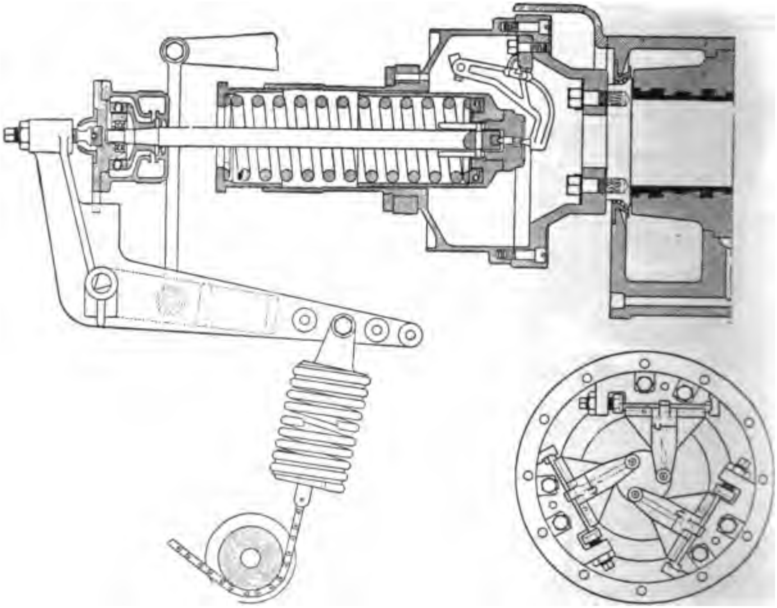


FIG. 2,501.—Ridgway high pressure turbine. ***In construction,*** the casing and heads are split horizontally. The diaphragms are also split, the top halves being fastened to the top of the casing and lift with it. The steam and exhaust connections are made to the lower half, and need not be disturbed when the turbine is opened. For high pressure bores and diaphragms, the packing is made of carbon blocks; for low pressure bores, a water impeller is used. A throttling governor (fig. 2,502) is used with the valves, directly connected. Depending on operating conditions, various combinations of valves are used. For high pressure service the main valve is usually of the double balanced poppet type, feeding an arc of nozzles sufficient to carry full load. A second valve of the piston type, mounted on the same stem supplies a separate set of nozzles for carrying overloads, but does not open until the main valve reaches full load position. The construction of these valves is such that the main valve can always close and shut off the entire steam supply, if for any reason the turbine exceed normal speed. An additional hand operated valve is furnished, if required, for non-condensing service, which by passes the first and sometimes the second stage, but keeps the turbine under full control of the governor. An automatic and hand operated over speed trip is provided, to guard against the the possibility of derangement in main governor system.

mixed with water, which might happen in cases of leakage from the turbine glands, forms an emulsion like jelly and chokes up the coil. If the oil turn a whitish yellow and do not seem to be well mixed, water leaks in the cooler may be looked for, or, when shut down, water leak from condensation through the water packing glands to the bearing. If the oil cooling system be independent of the water glands, it is advisable to leave the water slightly turned on at all times.

In the general care of the turbine, governor parts and connections with the primary and secondary valves must be regularly inspected to see that they do not become gummed; attention should also be given to the pilot valves to see that they work freely. It is advisable to use a light lubricating oil for these valves. A heavy oil will be found to be too sticky, and will deposit a soot around the piston, which makes the valves work stiffly when starting up, and even when heated through, has a bad effect on the good working of the valves. The pilot valves can easily be cleaned once a week by taking them out and removing any oil blackness. A little kerosene oil is admissable for this, but on no account should sand paper or emery paper be used, as this treatment will cause leaky valves. The grooves



FIG, 2,502.—Detail of Ridgeway high pressure turbine governor whose operation is described in fig. 2,501.

around the stem should be cleaned and any dirt or soot removed from the ends. Oil should not be used on these valve rods as they usually get sufficient lubrication.

If the primary valve does not seat properly, and leaks when closed, the seat may need grinding; ground glass and emery should be used for this, finishing with fine emery. Care should be taken to wipe the seats and discs well after grinding. If grinding do not remedy the defect the spring should be given more compression to close the valve. If it be found that the primary valve do not take care of the load properly, the bottom needle valve should be unscrewed to admit more steam to raise the valve.

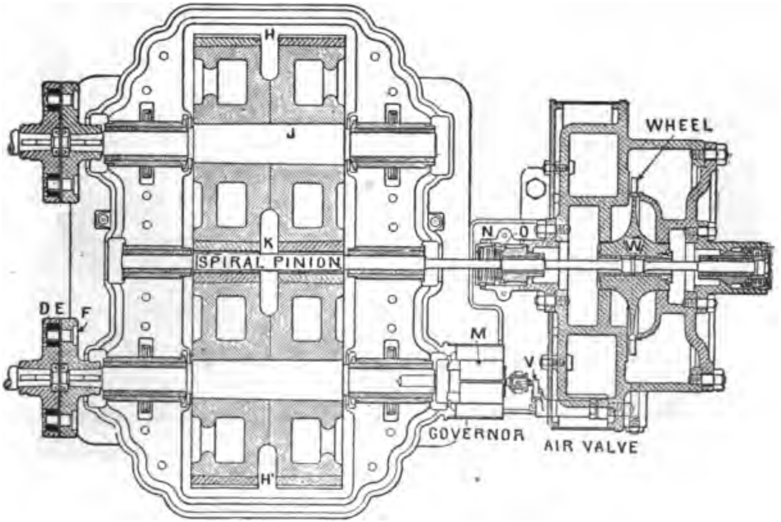


FIG. 2,503.—De Laval simple impulse turbine, consisting of a single wheel W, revolving at from 10,000 to 30,000 R.P.M. according to size, and which is attached to a long slender shaft allowing it to revolve about the center of gravity instead of its geometrical center. A set of spiral gears, K H H′, (ratio 10:1) reduces the speed and transmits power to two shafts, J J′, each being directly connected to a generator. The turbine bearings are held in place by springs N, which press against a collar O, formed like a socket, thus allowing flexure of the shaft. The couplings have pins F, driven into the holes, the opposite flanges being fitted with rubber bushings E, having internal steel bushings D, which slip over the end of pin F, to protect the rubber; this brings wear on the outside. The governor M, controls a throttle valve and in case of extreme speed opens a valve admitting air to the wheel compartment by lever V, the air friction checking the speed.

The top needle valve takes care only of the dash pot for an easy downward drop of the primary valve. The needle valves on the secondary valve work differently and rarely give trouble; the lower one admits steam which is cut off during exhaust by the pilot valve, and the upper one should be open enough to keep the valve from vibrating owing to puffs of steam within the turbine.

The blades and channels should be occasionally inspected and cleaned.

It is important that there be sufficient oil in the suction tank to avoid air being drawn into the system. The bearing pet cocks should be frequently tried to see if there be the proper circulation. The oil strainer in the base of the turbine should be removed and cleaned every few days.

Directions for Operating a De Laval Turbine.—In first starting, after erecting or long shut down, the bearings should be flooded with oil, the amount being gradually reduced to the normal quantity.

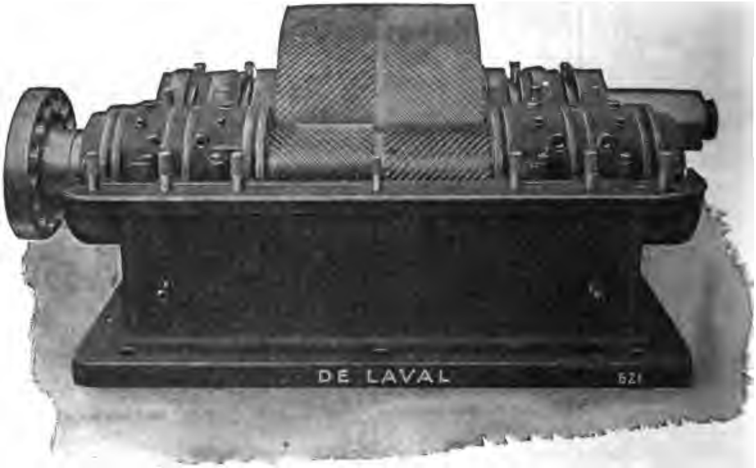


FIG. 2,504.—De Laval reduction gear, 500 horse power, 30,000 to 750 R. P. M. The reduction gear comprises a large gear, which is coupled to the shaft of the driven machine, and a pinion, which is coupled to the shaft of the turbine, or driving machine, the two being held accurately and rigidly at the proper center distance, and in alignment, by a heavy cast iron gear case. The main gear consists of a cast iron center or hub, upon which two seamless, rolled steel bands are shrunk and in which the teeth are cut. The center, or hub, is mounted on the gear shaft by means of a taper fit and key. After the complete assembling of the gear bands, center and shaft, the whole is carefully tested for static and running balance, after which the teeth are cut.

The oil reservoir on the self oiling bearings should contain oil in sufficient quantity to register between the rod marks on the gauge glass.

The small oil valves on the governor valve should be filled with cylinder oil, the valve stems then pressed down, thus allowing the oil to pass into the governor valve.

Steam may now be turned on, and the governor valve and wheel case allowed to become thoroughly heated.

Before doing this, the nozzle valves should be opened about a half turn, otherwise they will stick when the wheel case becomes hot.

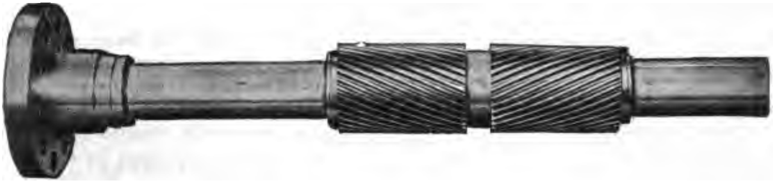
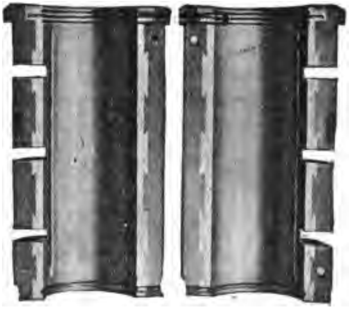


FIG. 2,505.—De Laval pinion. It is cut directly on the pinion shaft which is a nickel steel forging. All surfaces of the pinion with the exception of those of the teeth are ground; the pinion is tested for running and static balance.



FIGS. 2,506 and 2,507.—Reduction gearing bearings. The gear and pinion are each supported by two bearings, though in very large machines the pinions may be supported by three bearings. The bearing consists of cast iron shells, turned and babbitt lined. The shells are split for convenient removal without disturbing the shaft. To secure precision in alignment the extremes of the shells are ground to limit gauges. The caps are provided with oil catchers, and the shafts with "oil slingers" to prevent oil creeping out of the bearings. The pinion bearings are arranged for water cooling.

The turbine should be started gradually so as to give the bearings time to heat thoroughly; more time is required for this in the large turbines than in the smaller.

The self oiling bearings must be examined as soon as the turbine starts to see if the oil rings run properly.

If the turbine be running condensing, the condenser should be started first. When starting with no load, it is advisable to start with a low vacuum, say from 24 to 25 inches. As soon as the load is put on, the vacuum should be raised to its maximum.

In shutting down when running non-condensing, the throttle valve should be closed and the lubricator shut off as soon as the machine has come to a standstill.

If the turbine be running condensing, and if operating the water and air pumps, either directly or indirectly, the air cock on the exhaust end of the turbine wheel case should be opened before the throttle is closed.

In operating the usual precautions, with which engineers are familiar, should be taken to keep the lubrication system in working order.

The sight feed lubricator must be kept clean and the oil in the self oiling bearings and the gear case drawn off and filtered as often as necessary.

Particular attention should be given to the oiling of the governor mechanism and especially the contact surfaces between the governor pin and the plunger on the bell crank. The high speed bearings should be removed and examined at intervals.

In case a bearing run hot, it should be taken out, the oil grooves cleaned, and if any bright or black spots appear they should be removed with a scraper.

The strainer above the governor valve, which prevents foreign particles entering the turbine, should be removed and examined at least once a month.

If the turbine speed be too high, the brass nut holding the governor springs should be screwed out, or if the speed be too low, the nut should be tightened. It is well, every time a turbine is started, that the bell crank be pressed down to make sure that these parts do not stick; when fully depressed, the governor valve should shut off steam entirely, or at least within a few pounds.

The teeth of the gears should be cleaned occasionally when the machine is not running. Kerosene and a metal brush are best for this purpose.

The gear case should be cleaned at the same time and the gears well lubricated.

If, for any reason, the gears have to be taken out of the case, the engineer should secure special directions from the manufacturers relating to the adjustment as well as removal.

The life of the gear depends, to a considerable extent, upon being kept in perfect adjustment.

CHAPTER 45

INDICATORS

The indicator is an instrument used for the purpose of recording the pressure of the steam in the cylinder, at all points of the stroke, as the piston moves to and fro. This is done on a piece of paper secured to a revolving drum, by a pencil attached to the indicator piston.

An indicator consists of a small cylinder accurately bored out and fitted with a piston capable of working in the cylinder with little or no friction and yet practically steam tight; the piston rod is attached to a pair of light levers at the end of one of which is carried a pencil designed to move on a nearly up and down line.

The motion of the piston is controlled by a spring of known tension, several of which are furnished with each instrument; each spring is marked to show at what boiler pressure it is to be used. The elasticity of the spring is such that ***each pound pressure on the piston causes the pencil to move a certain fractional part of an inch.***

Attached to the instrument is a ***drum*** which has a diameter of about two inches, and around which is placed the paper, the ends passing underneath a piece of slit brass, fitted so that the paper can be held firmly after being wound around.

This cylinder is capable of a reciprocating or semi-rotative motion on its axis of such an extent that the extreme length of diagram may be about 5 inches.

NOTE.—The indicator is said to have been invented by James Watt, but it was at first vastly inferior in finish and accuracy to the improved forms now in use; these are all substantially of the same construction and act upon the same principle.

Fig. 2,508 is a sectional view of an indicator of the ***inside spring*** type. The part D, is a steam cylinder containing a piston. The part of the cylinder below the piston can be placed in connection with either end of the engine cylinder. Above the piston is a spiral spring. The piston rod is guided by a cover. The end of the rod P, is connected to the ***parallel movement*** consisting of two levers pivoted at A, at one end, and having pivoted at the other end a link which carries a pencil at its center. F, is the drum, and G, the drum pin on which the drum reciprocates. The indicator is piped to the cylinder so that by turning valves it can be connected with either end of the cylinder.

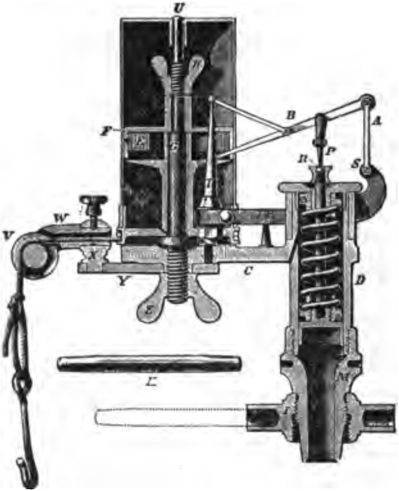
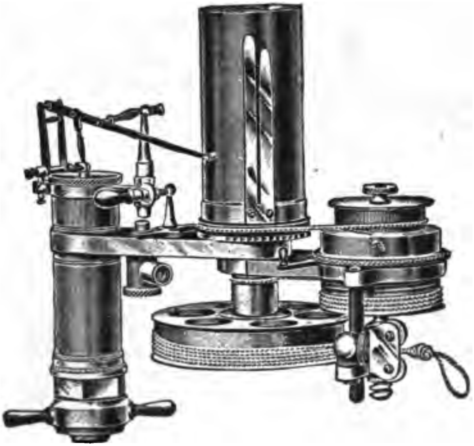


FIG. 2,508.—Sectional diagram of an indicator. A, is the swinging bar; B, the pencil bar; C, the indicator frame; D, cylinder containing tension spring; E, coiled spring on drum roller; F, revolving cylinder or drum; G, drum pin; H, and Z, thumb screws holding drum; I, nut to connect indicator to pipe; K, lever for screwing up I; M, and N (fig. 2,580), connection between the spring cylinder and pipe; O (fig. 2,580), specially small piston; P, piston rod; R, joint; S, pin; T, post for guide of pencil bar; V, guide pulley for cord from reducing lever; W, swivel sleeve for cord; X, swivel pin; Y, support for swivel pin.

The drum is made to reciprocate synchronously with the piston by driving it from the cross head through a suitable ***reducing motion.***

In operation, the steam pressure in the cylinder D, is the same as in the cylinder at every point of the stroke, hence the piston will move up and down as the pressure varies, and having placed a piece of paper or *card* around the drum, if the pencil be pressed against same, it will trace a *diagram* on the card and thus "indicate" accurately what is going on inside of the cylinder, viz:

1. The exact point of the stroke at which steam is admitted.
2. The initial pressure of the steam in the cylinders, which being compared to the boiler pressure, shows us whether the steam pipes and passages are of the necessary dimensions.
3. The way in which the initial pressure is maintained or otherwise during the period of admission.
4. The point of cut-off.
5. The pressure during the whole period of expansion.

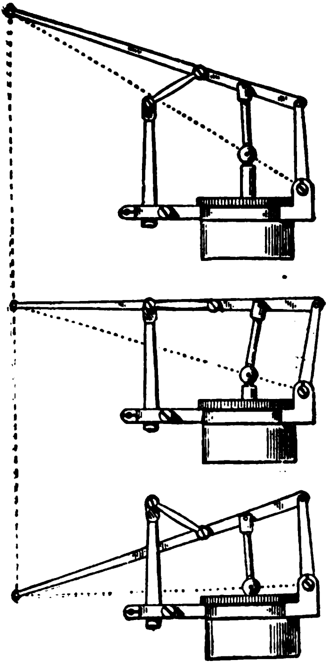


FIGS. 2,509.—Lippincott inside spring indicator with reducing wheel. The ½ inch cylinder and piston may be removed, and the ¼ inch cylinder substituted. With the ¼ inch piston, each spring becomes suitable for double the pressure allowable for the ½ inch piston. To illustrate:—with the ½ inch piston, the 60 lb. spring is suitable for 120 lbs. initial pressure. With the ¼ inch. piston, the 60 lb. spring is suitable for 240 lbs. pressure. The ¼ inch cylinder is made of special alloy, to withstand the action of *ammonia*, and is also suitable for gas engine and hydraulic work.

6. The point of pre-release, *i.e.* when the exhaust is opened.
7. The rapidity with which the exhaust takes place, as shown by the nature of the exhaust curve.
8. The minimum back pressure, which in a condensing engine is also the test of the perfection of the vacuum, and in a non-condensing engine shows what the effect of the friction of the exhaust pipes and passages is in addition to the unavoidable pressure due to the atmosphere.

9. The period when the exhaust is closed.
10. The nature of the curve of compression.
11. The power which is being developed by the engine, knowing the revolutions per minute.

Parallel Motion.—To impart a straight vertical motion or a motion "parallel to the vertical" upon the paper card wound around the drum, a *parallel motion* device is necessary. There are various forms of this movement, one being shown in fig. 2,508, and another, as used in the Thompson indicator, in figs. 2,510 to 2,512, which illustrates the vertical line traced by the pencil, the pencil lever being shown in its central and both extreme positions.



FIGS. 2,510 to 2,512.—Thompson parallel motion. It consists of a pencil lever pivoted at one end to a radial arm and at an intermediate point to a connecting arm, the action of the latter giving the parallel motion. Between the radial and connecting arms is pivoted a link having its other end connected to the piston rod. The parallel motion of the device is progressively shown in the three views.

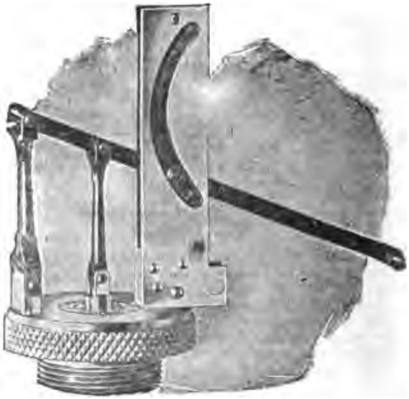
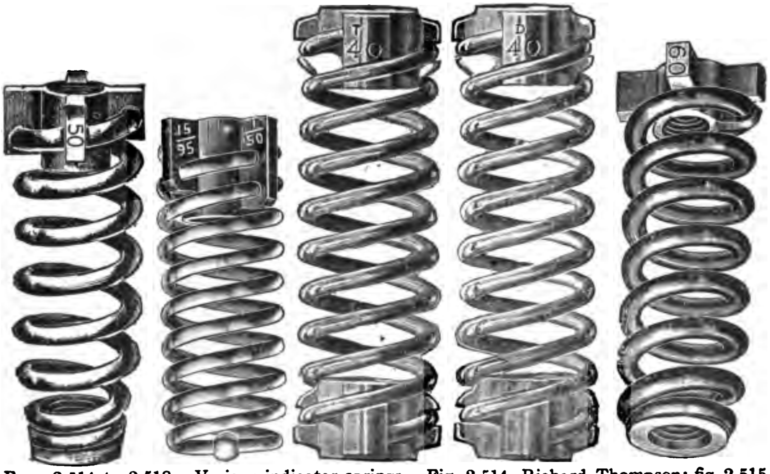


FIG. 2,513.—Ashcroft parallel motion. ***It consists of*** a vertical plate secured to the steam cylinder. In this plate is a curved slot in which works a roller revolving on a pin set in the pencil bar. The curve of the slot and the location of the roller on the pencil bar compensate for the tendency of the pencil to move in an arc, resulting in a travel of the pencil point parallel to that of the piston.

Indicator Springs.—The spring is the vital part of an indicator, since upon its accuracy depends that of the diagrams. Any variation from the true proportional compression is magnified many times by the pencil movement, and the greatest care is necessary in the manufacture and calibration of the spring. An indicator outfit usually contains several springs enabling the indicator to be used for various initial pressures.

The numbers by which *indicator springs* are designated denote the pounds pressure of steam required to raise the pencil one inch; *the corresponding scales* are divided so that each division represents one pound of steam.



FIGS. 2,514 to 2,518.—Various indicator springs. Fig. 2,514, Richard Thompson; fig. 2,515, Schaffer and Budenberg; 2,516 and 2,517, Tabor, duplex; 2,518 Trill.

Fig. 2,519 shows the kind of scale used; it has six faces giving six scales, the ones visible being 10 and 30. If a diagram be taken with a No. 30 spring, the No. 30 scale when placed upon it will give direct pressure readings as shown in fig. 2,520.

Preparing the Indicator for Use.—In selecting a spring aim to get as large a card as possible without undue distortion.

If a card be taken with say a 20 spring, an error of measurement of $\frac{1}{100}$ of an inch would influence the results only $1/5$ lb. With a 50 spring, the same error in measurement would represent a departure of ½ lb.

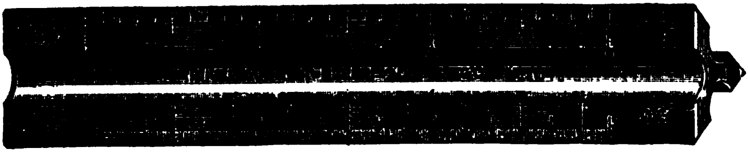


Fig. 2,519.—Triangular boxwood indicator scale. The six faces give six different scales corresponding to springs of like number.

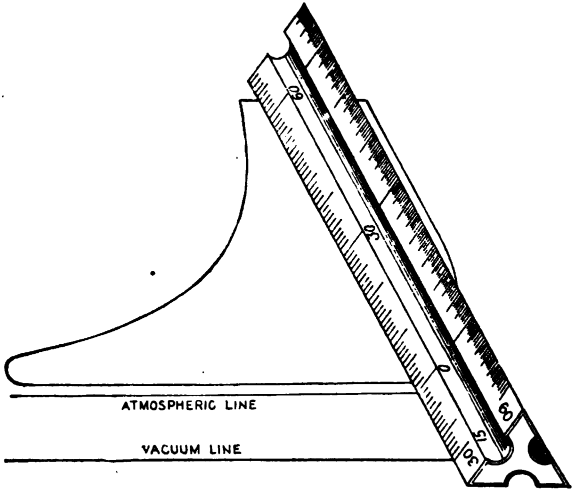


Fig. 2,520.—Triangular boxwood indicator scale as applied to a diagram showing that the pressure in the cylinder when the piston is at point A, of the stroke is, by measurement, 64 lbs. gauge pressure. If zero of the scale were placed in the vacuum line, the reading would be in pounds absolute.

Of course, the speed must be considered, as the allowable movement of both the pencil and drum is limited by the effects of momentum. At high speeds a light spring and lag movement of the drum would result in a diagram as distorted by the effects of momentum and inertia as to introduce errors more serious than those due to inaccurate spring measurement. The standard size of paper drum is used for moderate speeds, and the smaller drums which some of the makers supply for high speed work, these drums in some cases being interchangeably used upon the same instrument. Fig. 2,521 shows the method of changing a spring.

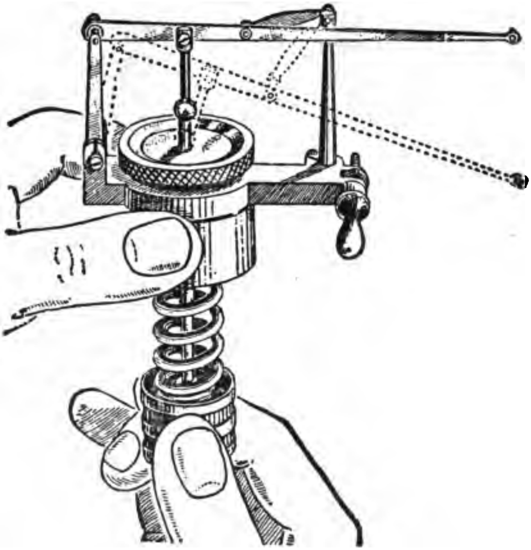
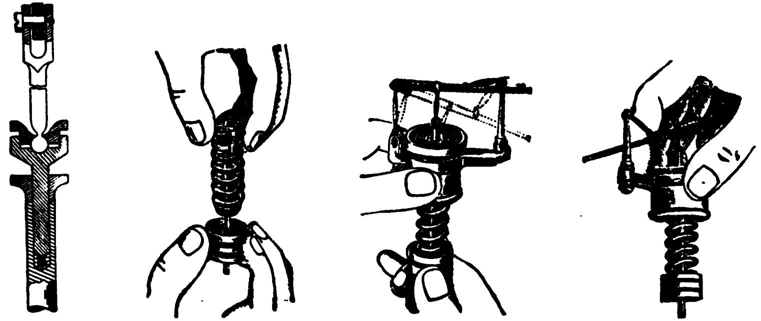


FIG. 2,521.—Head of Thompson indicator showing method of changing a spring. Unscrew the head of the indicator, hold the carrying ring as shown, and the piston and spring may be easily disconnected from the moving parts and head.

In some instruments the position of the atmospheric line is fixed; in others it is adjustable, so that in indicating a non-condensing engine the base line may be lowered, and the whole of the allowable movement of the pencil utilized for the height of the diagram.

The adjustment for height is effected by lengthening or shortening the distance between pencil lever and piston.

Select a hard lead of good smooth quality and of small diameter, and use



FIGS. 2,522 to 2,525.—Assembling of Trill cap, sleeve, and piston. These form one unit. ***In assembling,*** the moving head of the spring is fastened to the piston by five turns of the spring as in fig. 2,523. The piston rod is then placed in the cap, and the wing end of the spring is screwed to the cap, fig. 2,524. The piston, spring, and pencil movement complete are then ready to be put in place, which is done by pushing the sleeve down in the cylinder and screwing the cap down until it stops as in fig. 2,525. The swivel joint is now screwed into the upper end of the piston rod until the pencil point reaches the desired point on the card.

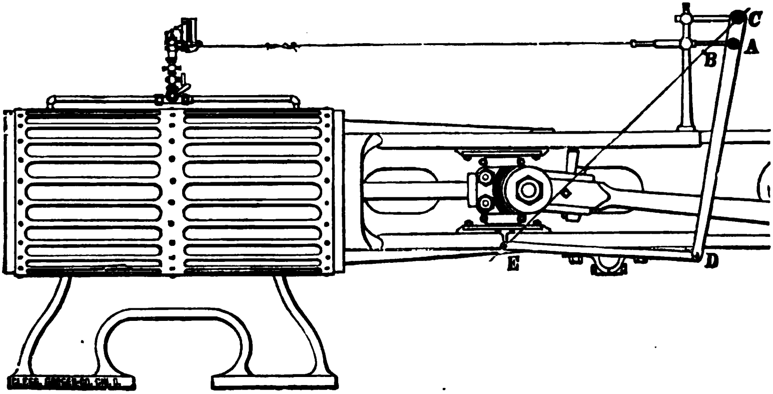


FIG. 2,526.—Lever reducing motion. CD, is the lever which is pivoted at one end to a fixed point C, and at the other D, to a link DE, which is pivoted at E, to the cross head. The cord for the indicator drum is attached to the lever CD, at same point A, so selected as to give the proper length of movement to the drum. A spring attached to the drum keeps the card taut at all times. The length of the link DE, should be such that CD, will be vertical when the cross head is at half stroke. In all lever motions there is a radical defect due to the fact that while the cross head moves in a straight line, any point on the lever swings through an arc of a circle.

only a small piece at a time. At the end of the pencil lever, where the motion is greatest, the weight should be reduced to the smallest possible value. If pointed with a fine file and rubbed down with an emery stick, such as is used for sharpening draftsmen's pencils, or a fine stone, it will wear longer and be smoother and more satisfactory than if whittled into shape.

For lubricating the bearings of the instrument a light machinery oil should be used, one that will not gum or corrode. A small vial of such oil usually accompanies the instrument, some makers furnishing porpoise oil, such as is used for clocks and watches. The piston, however, is better lubricated with cylinder oil, which must be absolutely free of grit.

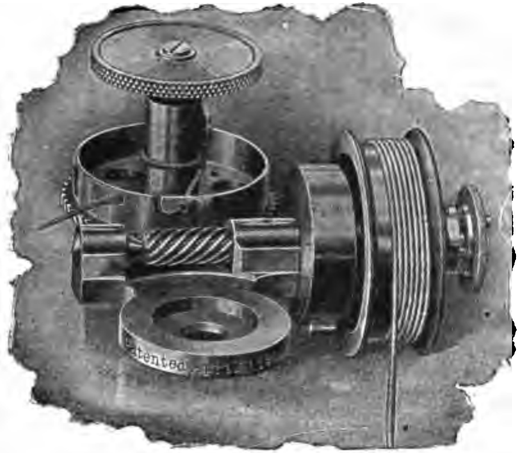


FIG. 2,527.—Ashcroft reducing wheel. ***The construction*** of this device is similar to that of the Houghtaling reducing motion. The base is extended to provide a support for the worm gear disc to which the cord for the indicator drum is secured. In other respects this device is the same as the Houghtaling gear except that the knurled thumb piece on the worm gear disc corresponds to the knurled thumb piece on the top of the indicator paper drum, when the Houghtaling gear is employed.

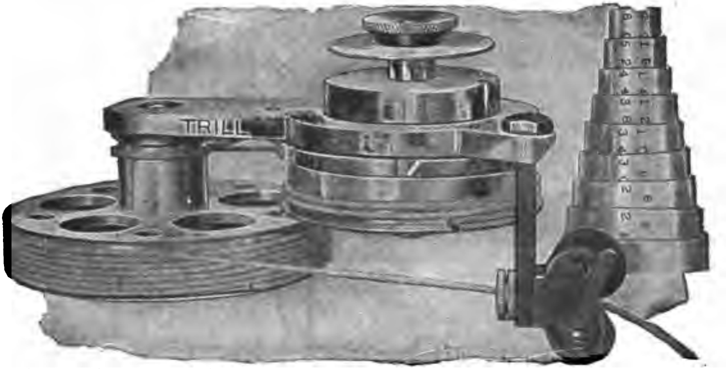


FIG. 2,523.—Trill reducing wheel with nine additional bushings giving 18 reductions, covering strokes from 6 ins. to 5 ft., with cord hooked direct to the cross head.

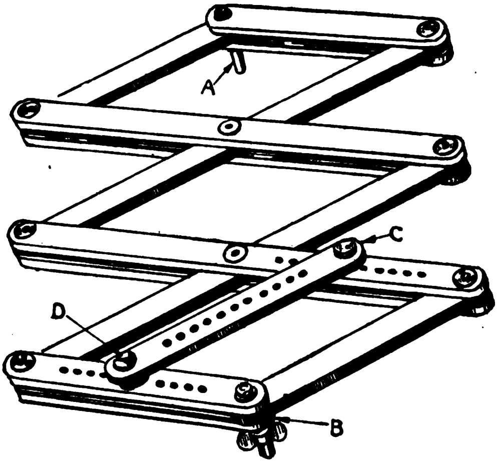


Fig. 2,529.—Pantograph reducing motion. This device is easily made. The members usually consist of strips of hardwood 1⅛ × $\frac{5}{16}$ × 16 ins. The pantograph is pivoted at the point B, by a stud or winged thumb nut to a part of any character secured to the engine room floor, and the nut A, is adjusted to a suitable piece secured to the cross head of the engine. The driving cord for the indicator is attached at the pin E, the position of which may be changed by placing the pins C and D, in the holes provided for them. In doing this, the pin E, must always be on the straight line between the points A and B.

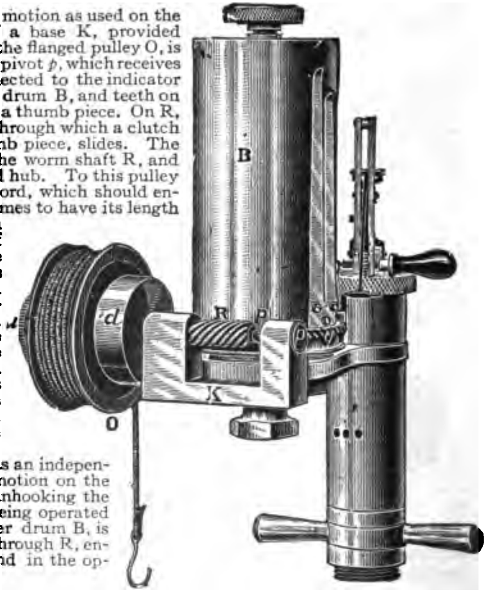
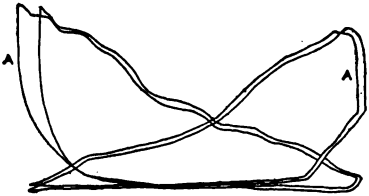


Fig. 2,530.—Houghtaling reducing motion as used on the Tabor indicator. It consists of a base K, provided with a worm shaft R, on which the flanged pulley O, is rotated, the outer bearing being a pivot *p*, which receives the thrust of shaft R. It is connected to the indicator upon the arm that supports the drum B, and teeth on spool *g*. *d* is a spring case, and *u*, a thumb piece. On R, is secured a collar (not shown) through which a clutch pin, secured direct to the thumb piece, slides. The flanged pulley O, runs freely on the worm shaft R, and has on its outside a clutch shaped hub. To this pulley O, is connected the actuating cord, which should encircle it a sufficient number of times to have its length when unwound a little more than equal the length of the stroke of the engine. The other end of the cord is secured either to the cross head of the engine, to a standard bolted to the same, or to any moving part that has a similar motion, and must be connected in line from pulley O. Enclosed in the spring case *d*, is a small, plain spiral steel spring which operates to return the pulley O, back to its starting point, after it has been revolved in one direction by the forward movement of the engine cross head. As this pulley O, has an independent, rotating back and forth motion on the worm shaft R, the necessity of unhooking the cord when the indicator is not being operated is entirely overcome. The paper drum B, is rotated forward by the pulley O, through R, engaging with the worm gear *g*, and in the opposite direction by the spring.

Reducing Motions.—In order to use the indicator, the drum must move in step with the piston. The movement is usually derived from the cross head, and the appliance used to *reduce* the movement to that adapted to the drum is called the *reducing motion.* The accompanying cuts show the various types of reducing motion in general use. Fig. 2,526 shows the ordinary lever motion, fig. 2,529, the pantograph, and fig. 2,509 a reducing wheel.



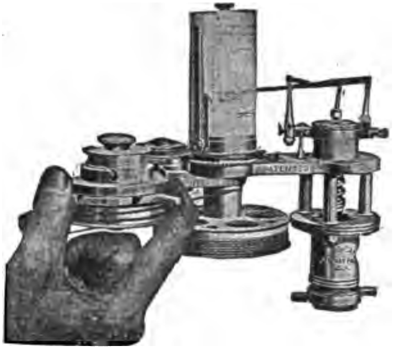


FIG. 2,531.—Error due to cord whipping. The failure to recoil promptly on the return stroke will permit the cord to slacken, and possibly become entangled in the wheels with probable injury to the instrument. The tension or pull of the recoil spring should be just sufficient to prevent whipping. Obviously high speeds will require more tension than low speeds, hence, a tension adjustment is a desirable feature. The jerking or unequal pull of the cord also has its effect in recording an error (wavy lines) on the card as shown above.

FIG. 2,532.—Adjusting the tension of the Trill indicator recoil spring. *In construction,* a disc is placed on the same shaft with the nest of bushings and wheel enclosing the recoil spring. On the edge of the disc are notches for the reception of a pencil located under the frame arm. The inner end of the recoil spring is fastened to the disc, so that by turning the latter, as shown, the pull of the recoil spring is increased to any desired tension, and by removing the panel and letting the spring unwind, the opposite result is obtained.

Indicator Piping.—The connection between the indicator and the engine cylinder must be as direct as possible, so that the pressure acting upon the indicator piston at any instant will be as nearly as possible the same as the pressure acting upon the engine piston at the same instant.

All modern engines are tapped for the indicator, but if there

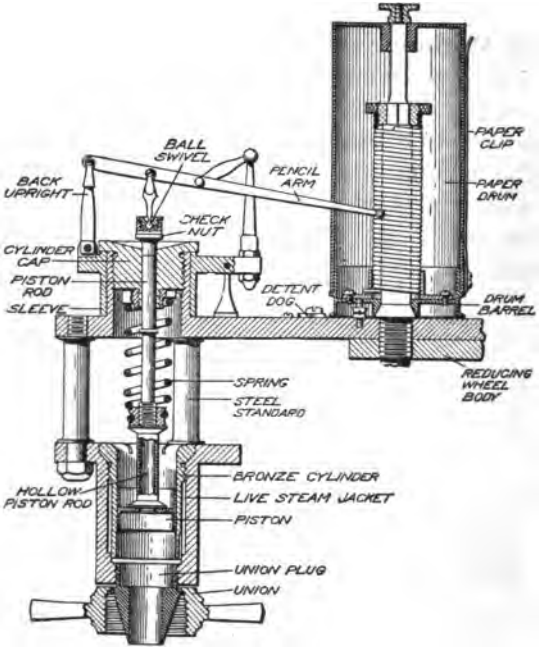
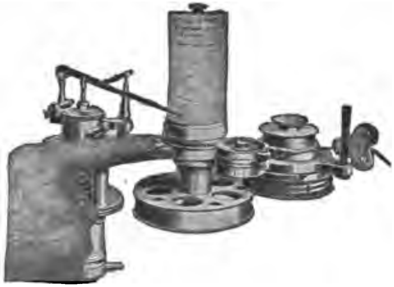


FIG. 2,533.—Trill outside spring indicator; cross section showing principal parts.



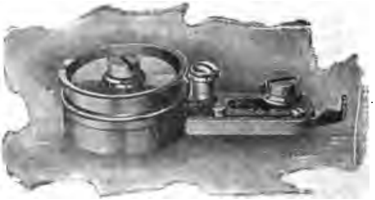


FIG. 2,534.—Stopping the paper drum of Trill indicator. Cord connection for engine equupped with a reducing motion.

FIG. 2,535.—Trill cord take up. With this device the paper drum of the indicator can be stopped for removing the card and replacing the paper without disconnecting the cord from the cross head. Its purpose is to take up the cord motion when the drum is stopped and keep the cord taut between the reducing wheel and cross head.

should be no such provision, the holes should be drilled in the counter bore of the cylinder, and tapped for a ½ inch pipe thread.

If a single indicator be employed, the connection piping is usually made as in fig. 2,536. When this connection is used, the bends should be as easy as possible, the ordinary elbows are not suitable.

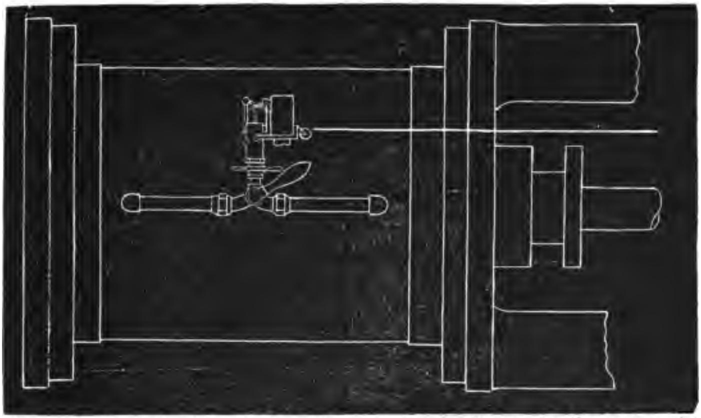


Fig. 2,536.—Indicator piped to cylinder with two way cock. By turning the cock to right or left the indicator is put in communication with either end of the cylinder, thus enabling diagrams to be taken from both head and crank ends without disconnecting the indicator.

Figs. 2,537 and 2,538.—Two types of cock used in indicator piping. Fig. 2,537 two way cock; fig. 2,538, long turn angle cock. The two way cock is piped as shown in fig. 2,536. However, since the piping is usually permanent, this method materially increases the clearance of the engine which is objectionable, and is reduced by substituting for the two way cock an easy turn T and, using two angle cocks in place of the elbows.

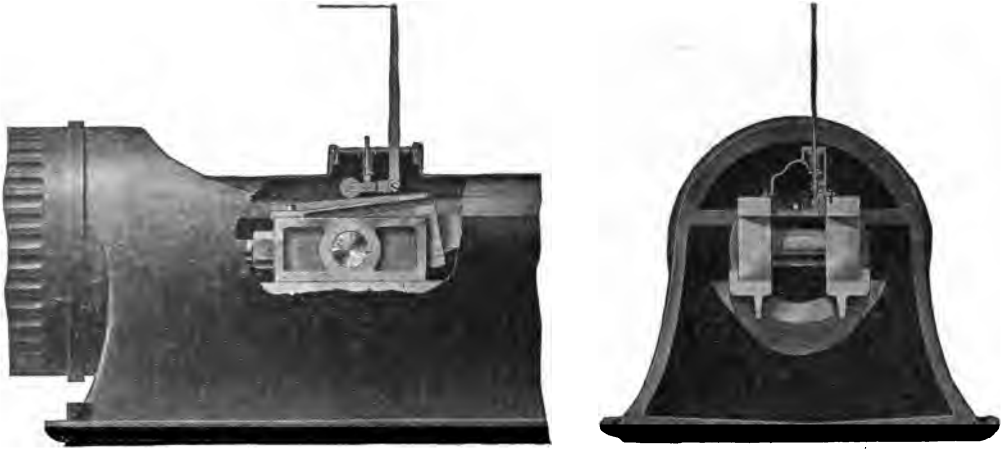


FIG. 2,539.—American-Ball engine reducing motion. ***In construction,*** the cross head is provided with an inclined plane surface adapting to imparting lateral motion to piece which contacts with the inclined plane surface. A bell crank carrying a wheel on the forked end of the short arm is pivoted so that the wheel may be held in contact with the inclined plane of cross head, and when the device is in action a spring is arranged to maintain contact. The long arm of the bell crank lever is arranged for attaching the cord leading to indicator, and the location of the pivot is such that the pull of the cord also helps to maintain the contact with the inclined plane. ***In stopping*** the mechanism, a trigger is arranged to engage a notch in the side of a rod attached to the bell crank, the location of the notch being such that the wheel just clears the highest point of the inclined plane when the trigger enters the notch. To engage this trigger, the bell crank is pushed slightly beyond its normal swing, so that the trigger may enter the notch and thus prevent any further motion. When ready to take another card the trigger is disengaged and the reducing motion is instantly in service. The cord remains attached to the arm, and the whole outfit, including the indicator drum, is started and stopped by the trigger.

How to take an Indicator Card.—When the instrument is properly connected to the engine, open the cock and let in the steam, which will set in motion the piston and levers. Press one finger lightly on top of the piston rod of the instrument to see if it be working smoothly.

If a rough action be felt, indicating the presence of grit or some derangement, shut the steam cock and correct the fault. The paper is put on the drum by wrapping it snugly around the drum at the top, bending it around and allowing the ends to project between the clips at the top, then by taking the lower corners which protrude between the thumb and the second finger, the paper is drawn down tightly over the drum. Now open steam cock and warm instrument.

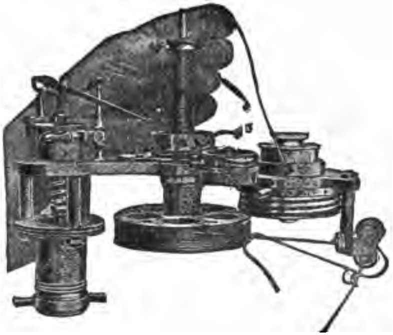


FIG. 2,540.—Replacement of intermediate cords between drum and reducing wheel of Trill indicator. To replace, cut cord to proper length, knot the ends and slip into the slotted holes.

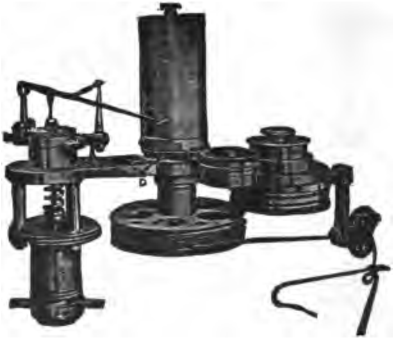


FIG. 2,541.—Treatment of main cord. Should the main cord slack and fall off of large wheel, wrap it on the wheel in any manner.

On non-condensing engines it is well to turn the cock so that the steam will blow into the atmosphere until it shows blue and dry. When the water has disappeared and the pencil is vibrating smoothly, the paper drum being in motion, hold the pencil lightly against the paper and allow it to trace the diagram.

For ordinary purposes of exhibition, showing the valve action, distribution, etc., one revolution is sufficient to hold the pencil in contact with the paper.

To show the governor action, variation of load, etc., the pencil will have to be held on for a number of revolutions; and when measuring power, the pencil should be allowed to pass from ten to twenty times over, and the average diagram measured.

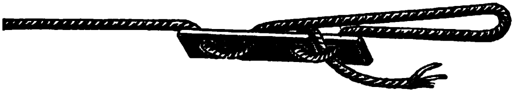


FIG. 2,542.—Connecting cord and method of attaching. The indicator should be connected to the engine cross head by as short a length of cord as possible. Cord having very little stretch, such as accompanies the instrument, should be used; and in cases of very long lengths wire should be used. The short piece of cord connected with the indicator is usually provided with a hook; and at the end of the cord, connected with the engine, a running loop can be made, by means of the small plate sent with each instrument, in the manner shown in the accomanying cut, by which the cord can be adjusted to the proper length, and lengthened or shortened as required.

Turn the cock off and bring the pencil again to the paper tracing the atmospheric line. It is not good practice to trace the atmospheric line first, as the indicator and spring are not then heated, and under the same conditions as when the diagram is taken.

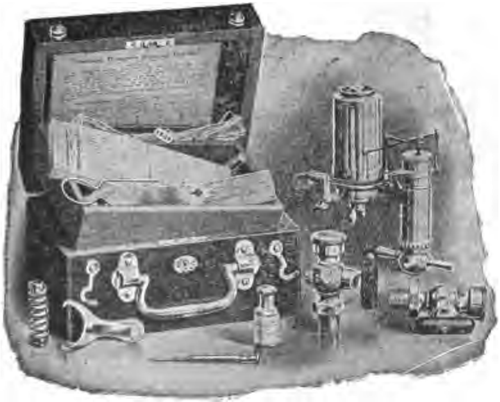


FIG. 2,543.—American Thompson indicator outfit consisting of indicator, two springs, two single cocks, one scale, wrenches, screw driver, porpoise oil, cards and instructions, enclosed in mahogany case. This represents the usual articles comprising an indicator outfit.

If there be more than one card to be taken, it is necessary to stop the drum while removing the card from the clips; this is done in some cases by making the cord in two lengths, and unhooking, whenever the drum is to be stopped.

The Indicator Diagram.—A diagram such as shown in fig. 2,544 would be traced by the indicator pencil, if the action of all the different operations should be perfect.

This, however, does not occur in practice, as it takes time for the valve to open, and to cut off the steam: thus the corners of the diagram cannot be as sharp as shown in the figure; also

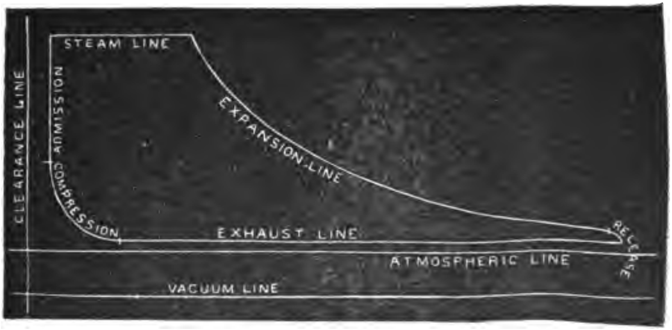


Fig. 2,544.—Theoretical indicator diagram illustrating the various "lines" referred to in the accompanying text. The tracing of an indicator diagram is effected in the following manner: Steam enters the indicator cylinder, and, forcing the pencil upward on the drum, causes it to make the ***admission line.*** This happens at the beginning of the stroke. The cross head now begins to move outward, revolving the drum, while the pencil is held up by the steam pressure; the ***steam line*** is thus drawn until the point of cut off is reached, when the valve cuts off the steam supply. As the cross head still continues to move, the steam beyond the piston is expanded and its pressure reduced, which allows the indicator spring, previously held under tension, to force the pencil downward; but as, on account of the motion of the drum it cannot descend in a straight line, it traces a curve, which records the steam pressure at all different points of the stroke after cut off. This curve is termed the ***expansion line.*** When the cross head nearly reaches the end of the stroke, pre-release takes place, that is, the exhaust valve opens, causing the steam to rush out of the cylinder and the indicator pencil to drop, making the ***pre-release line.*** The cross head now begins on its return stroke, and the drum spring revolves the drum in the opposite direction. The pencil, being at a height corresponding with the exhaust pressure, now marks the ***exhaust line,*** which may be either above or below the atmospheric line, according to whether the engine be running with back pressure or condensing. At a point near the end of the exhaust stroke the exhaust valve closes; and the remaining steam in the cylinder, having no passage to escape, is compressed by the advancing piston, thus raising the pressure, and also the indicator pencil, which now draws the ***compression line.***

the expansion line can never correspond with the adiabatic curve on account of *condensation* and *re-evaporation.*

The outline of an indicator diagram represents six different operations, which are indicated by different "lines" made by the pencil, and are partly due to the steam pressure acting upon the indicator piston, and partly due to the motion of the cross head and the corresponding rotation of the drum. These lines are called

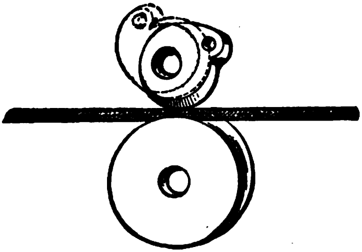


FIG. 2,545.—Trill cord carrier pulley and detent. The pulley is mounted on a hollow swivel post, by which the cord may be made to lead from the indicator in any direction. For indicating short stroke engines where the upper wheels of the reducing motion are used, the guide pulley is adjusted as shown in fig. 2,534. For longer strokes using the lower hole, it is projected below to lead to the larger wheel as shown in fig. 2,540. The purpose of the eccentric disc cord clamp is to secure the cord when wishing to remove the hook from the cross head. When the eccentric cam is thrown down into the position shown, the cord is free to draw out, without jerk on the instrument, but as the cord is held taut and cannot return, the hook drops off. Also when removing the instrument from the engine, it holds the cord under sufficient tension to keep it in place.

1. Admission line;
2. Steam line;
3. Expansion line;
4. Pre-release line;
5. Release or exhaust line;
6. Compression line.

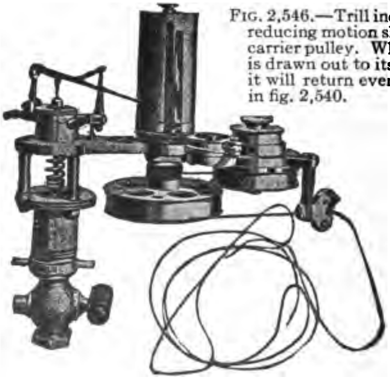
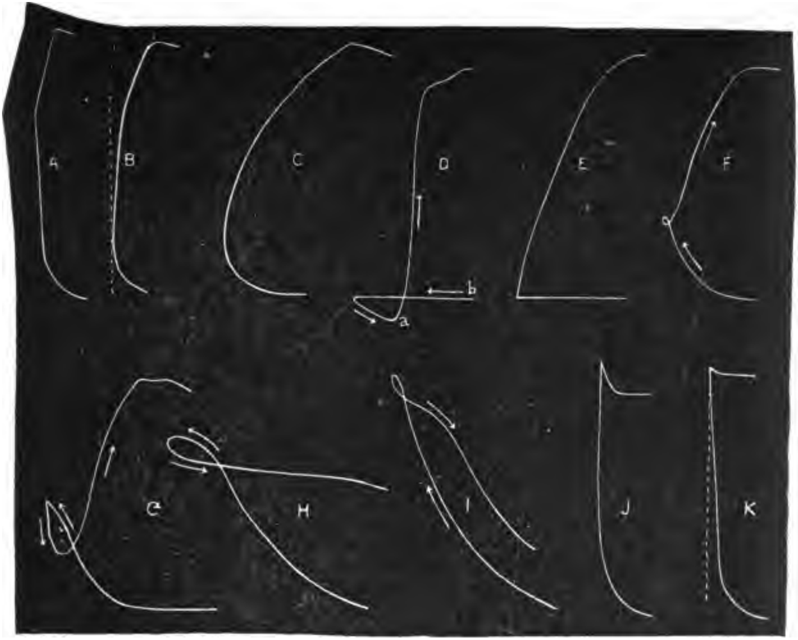
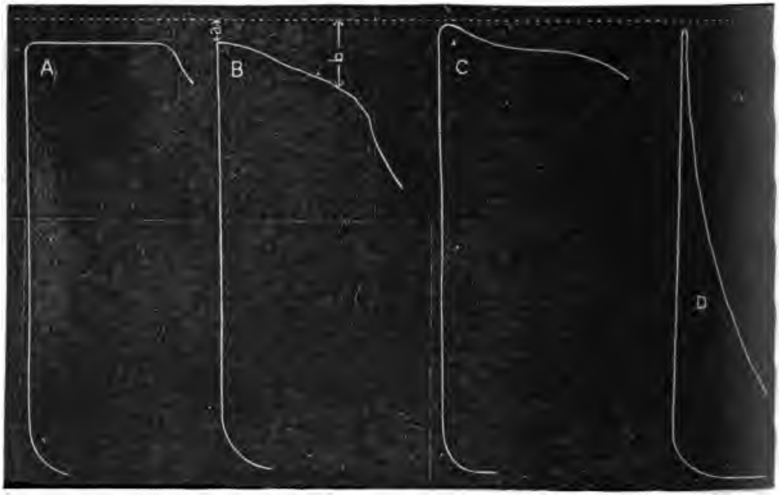


FIG. 2,546.—Trill indicator with reducing motion showing cord carrier pulley. When the cord is drawn out to its full length, it will return evenly coiled as in fig. 2,540.



FIGS. 2,547 to 2,557.—***The admission line.*** At A, is shown a normal or theoretical line. ***In practice,*** in order for the admission line to conform to A, it is necessary that the steam valve shall be open so as to admit the full pressure before the piston begins to move away; and this involves the question of lead, or the amount of opening which the valve has when the engine is on the center, and which, for many reasons, it is desirable to keep as small as possible, and yet allow the admission line to be perpendicular. As the steam valve is allowed to become late in opening, and the piston gets into motion before the steam is admitted, the admission line begins to curve inward, as at B and C, the leaning tendency increasing as the line progresses and the motion of the piston becomes faster. At D, is shown a peculiar admission line on a diagram of a slide valve engine, the eccentric of which had slipped so as to make the whole valve motion late. The exhaust closure being late as well as the steam opening, the compression was entirely cut out, and the back pressure line *b*, continued straight up to the end of the stroke. When the piston commenced its return stroke the steam valve had not opened. The exhaust valve had by that time closed the space between the cylinder head and the retreating piston was shut in, and as the piston moved away, a vacuum was created, running the pressure down toward *a*, as is shown by the arrow. At *a*, the steam was admitted suddenly and the admission line ran up, leaving the loop on the heel of the diagram as shown. The admission line may lean in, however, from another cause than that of the steam valves being late, as at E. The natural inference from the appearance of the diagram would be that the engine was late all around, but the fact is that the steam valve has plenty of lead, and therefore opens before the return stroke is completed. The exhaust valve is so late that it not only does not close for compression, but does not close until the piston is well started on the forward stroke, so that the steam is blowing through into exhaust, and the pressure drops. As the exhaust closes, the pressure increases, but the piston is moving away



FIGS. 2,558 to 2,561.—***The steam line.*** A, represents a good steam line. When the connecting pipe and passages are small for the piston speed and diameter, the steam line falls away as at B, the difference between the beginning of the stroke, and a point near cut off being shown by *a* and *b*. The steam line shown at C, is often met with on engines having a large steam chest and small steam pipe, the steam chest in this case acting as a reservoir, and allowing the steam in the cylinder to almost equal boiler pressure at the commencement of the stroke; but if the steam pipe be small, this pressure cannot be kept up when the piston is advancing. Diagrams are sometimes met with which have no steam line, the load being so light that the expansion of the steam in the clearance is sufficient to keep the engine in motion. In this case the expansion line meets the admission line at a point, as at D. The shape of the steam line is often modified by the admission, and it will be readily understood that it is difficult to say when the one leaves off and the other begins, under frequently occurring conditions.

FIGS. 2,547 to 2,557.—*Text continued.*

rapidly and the line never becomes erect. The degree of compression has much to do with the appearance of the admission line. The effect shown at F, is a very common one, produced by the pressure running up by compression to the point and falling away as the piston starts back before the steam valve opens, forming the loop. A more aggravated case of the same action is shown at G. This loop assumes all sorts of forms, according to the relations of the compression and admission, and the proportions of the openings and the piston speed; and it may even form when the steam valve opens promptly, by excessive compression, as is frequently seen on diagrams from the ordinary type of single valve, high speed engines with shaft governors, where the compression is increased as the load diminishes, resulting in admission lines like those shown at H and I. In the first of these the pressure is so low that the compression line extends above it, and when the steam valve opens, there is an escape of steam from the cylinder and the pressure is lowered to that at which the steam will flow from the chest. The appearance at I, is produced when the engine is lightly loaded, so that the compression is considerable. Just as a tardy action of the steam valve results in producing an inward leaning of the admission line, so a too early opening of that valve will result in the production of a line which leans outward, as shown at K. Any engine that is in line and properly adjusted in the connections, should run at the speed for which it is designed better with enough lead to bring the admission line upright than it does with more. A sharp point at the top of the admission line is usually an indication of too much lead, and it will be found to result in smoother running if the corner is just given an indication of rounding, as at A. The projection is due to the fling of the moving parts carrying the pencil above the point due to the pressure.

To these may be added the atmospheric line and vacuum or zero pressure line.

The Admission Line.—This line shows the manner in which steam is admitted to the cylinder. Under normal conditions, admission takes place suddenly while the piston is practically standing still at the end of the stroke, resulting in a straight line perpendicular to the atmospheric line, into which the compression line merges as shown at A, fig. 2,547. In practice the line usually departs from this, being variously distorted as shown in the figures.

The Steam Line.—This line indicates ***what percentage of the boiler pressure is realized in the cylinder, and how well it is maintained up to the point of cut off.***

In a really good diagram the steam line will appear about as at A, fig. 2,558, approaching, in its height above the atmospheric line, the distance indicated by the boiler pressure laid off to the same scale as that of the spring with which the diagram is taken, as shown by the dotted line, and remaining horizontal, or very nearly so, up to the point of cut off.

Figs. 2,559 to 2,561 show some of the usual distortions of the steam line.

The Expansion Line.—In all engines in which any pretention is made to economy, steam is used expansively. The expansion curve always varies from the theoretical or adiabatic curve due to condensation, re-evaporation and other conditions of operation.

A gas in expanding varies in pressure inversely as its volume; and steam follows this law with sufficient accuracy to make its application to the indicator diagram and to engine practice of value.

If the steam should be a gas, whose volume is not affected by the temperature of its surroundings, this law would hold true; but as the steam upon entering the cylinder will condense, its pressure falls more rapidly than due to expansion, until a point is reached where the temperature of the steam equals the temperature of the cylinder walls, and from thence the condensed steam will be re-evaporated, because the metal of the cylinder is hotter than the steam. Thus it will be understood, that the true expansion curve never equals the theoretical curve.

It falls below the theoretical curve at the first stages of expansion, and at a later stage, usually above it, crossing it at some point in the stroke.

The steam in the clearance space will affect the expansion, as it is filled with steam during admission, and all this steam expands with the steam that

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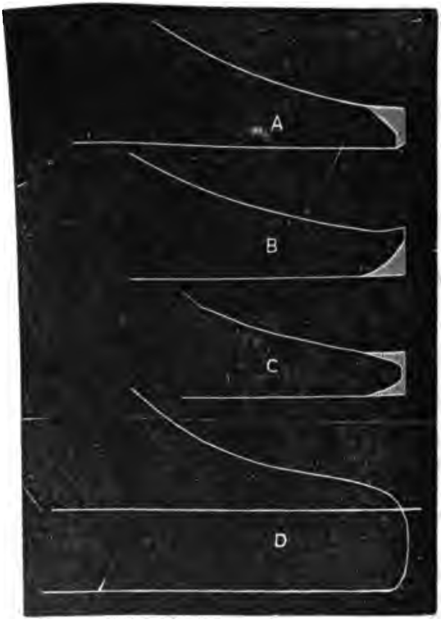
NOTE.—It is impossible to maintain in the cylinder the same pressure that is carried in the boiler, although with short connections, ample passages, and low piston speeds a very large percentage can be realized.



FIGS. 2,562 to 2,569.—***The pre-release line.*** The proper appearance of the pre-release line would be as at A, the pre-release occurring early enough to allow the pressure to fall nearly or quite to the line of back pressure by the time the end of the stroke is reached. If pre-release be delayed until the end of the stroke the appearance will be more like that indicated at B. If the pressure could be carried to the end of the stroke and immediately reduced to the line of back pressure, as indicated by the outline of the shaded space, it would be advisable to retain the full area: but since some area must be lost here in expelling the exhaust, it is better that it should be above the diagram at A, than below as at B. When the piston is approaching the end of its stroke, it has come to be a question of stopping it and sending it in the other direction. To do this smoothly, compression is applied on the other side of the piston, and obviously there is no object in keeping up the forward pressure, as at B. It is therefore better to let the pressure fall off, as at A, assisting instead of opposing the compression in bringing the moving parts quietly to rest, and by this early pre-release removing the back pressure represented by the shaded portion at B, so that the piston encounters less resistance in starting upon its bcakward stroke when it is an object to get it in motion. The difficulty of attaining this result on most engines is that where the lap is removed from a valve to cause it to open early and give an early release, the very lack of lap retards the closure and does not give sufficient compression. On the Corliss valve this may be corrected by setting the eccentric ahead, making both release and compression earlier; but disadvantages attend upon too great an angular advance of the eccentric, in the way of shortening the range of cut off, and the advantages of the valve motion in quick movement at admission, so that it is often necessary to divide the difference and compromise upon a point like that shown at C. The benefit of an early release is very apparent when a condenser is used; for, with an early release and a prompt realization of the vacuum, as at D, the largest possible percentage of the load is thrown upon the condenser, while a tardy release and a dragging action of the steam in leaving the cylinder results in the loss of a large area in the vacuum portion of the diagram as shown by the shaded portion of E, calling for a filled the cylinder volume before cut off, without the clearance volume being changed during expansion.

The expansion curve is often found to be wavy, a fact generally due to the indicator piston fitting too tight, which will allow it to bind, thus working in jerks instead of steadily.

The Pre-release Line.—By the pre-release line is understood the line beginning at the point of pre-release and extending to the end of the diagram. In practice steam must be pre-released just before the end of the stroke, in order to avoid excess back pressure. Figs. 2,562 to 2,569 show some examples of pre-release line.

The Exhaust Line.—The forward pressure upon the piston during the forward stroke is represented by the steam, expansion, and release lines;

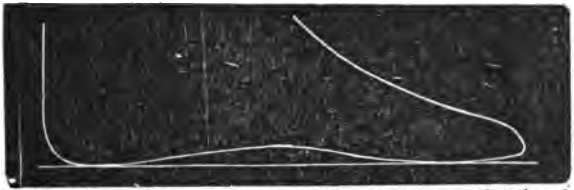


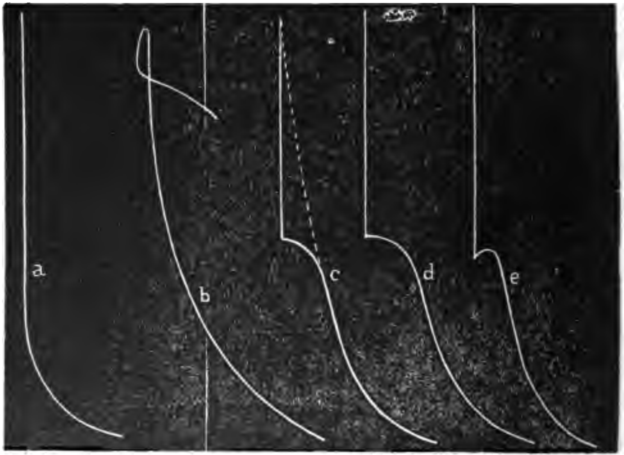
FIG. 2,570.—*The exhaust line.* The compression of the steam by the piston pushing it out of the cylinder against the resistance due to friction in pipes and passages, will show on the diagram in raising the line of counter pressure above the atmospheric line. In a well proportioned engine at moderate piston speeds, and exhausting through a short and ample exhaust pipe, this moving pressure will not be noticeable with an ordinary spring, and the line of back pressure will merge into the atmospheric line. In less advantageous circumstances, however, the back pressure line will be elevated above the atmospheric line. Sometimes a card is found where the back pressure line starts in well enough, but makes a gradual rise toward the center of the diagram, falling again as the stroke is completed, as shown. This is caused by too much inside lap on a slide valve narrowing up the exhaust passage as the center of the stroke is reached, where the piston, and consequently the steam, has the greatest velocity. The same effect may be produced upon a Corliss engine. It is also found where a pair of cylinders working on cranks set at 90° exhaust into the same pipe, the release of one cylinder occurring practically in the middle of the stroke of the other and the efflux of steam into the pipe causing a rise of pressure.

FIGS. 2,562 to 2,569—*Text continued.*

later cut off and more steam. When the cut off is late, more steam is admitted and has to be expelled; the appearance will then be more like G. Between this and the point shown at F, there may be any variety of shapes according to the terminal pressure and setting of the valves. When the steam is cut off so early that the expansion extends below atmospheric pressure, or pressure against which the engine is exhausting, the release line will be like that shown at H. Here when the exhaust valve opens, the pressure in the exhaust pipe is greater than that in the cylinder and when the valve is open at *a*, there is an inrush of the previously exhausted steam, raising the pressure to the back pressure line. This condition is apt to cause a disagreeable slamming of the exhaust valve, which is lifted from its seat when the pressure in the cylinder becomes less than that beneath the valve, and is slammed closed again when steam is admitted. It may be stopped by throttling the initial pressure so that the lessened expansion does not cause a loop. During the formation of this loop the pressure urging the piston forward has been less than that against which the piston moves ,the forward motion continuing only by reason of the momentum of the fly wheel and moving parts, so that the area of the loop represents just so much work exerted against the piston, and must be subtracted from the other area of the diagram to get at the effective work.

the pressure in the same end of the cylinder during the backward stroke is represented by the exhaust, compression, and admission lines.

If at the end of the stroke the steam has been expanded to atmospheric pressure in a non-condensing engine, there will be no immediate outrush of steam from the cylinder, because there is no greater pressure in the cylinder than that of the atmosphere into which the steam must flow. The steam must therefore be pushed out by the piston, and the resistance to its movement will depend upon the length and directness of the exhaust pipe, as well as its size. The general appearance of the exhaust line, is shown in fig. 2,570.



FIGS. 2,571 to 2,575.—***The compression line.*** At *a*, is shown a typical Corliss compression line, compression to about ⅙ initial pressure. At *b*, is shown excessive compression, the pressure running up above that in the steam chest, so that when the valve opens for admission, steam flows from the cylinder to the chest and the pressure falls. A form of compression line often met with is shown at *c*, where the pressure, instead of continuing upward along the dotted curve, falls away as shown. When this occurs you must look for some cause for the reduction of pressure, and you will generally find it in a leak. As the piston approaches the end of its stroke its movement becomes very slow. The volume of steam involved is small and growing smaller, and if there be even a slight leak in the exhaust valve, drip valve, or piston there will come a time when the volume of steam discharged through the leak will equal the volume generated by the movement of the piston in the same time. If the pressure of the outrushing steam be kept constant by the advancing piston, the line will have the appearance as shown at *d*. If the leak should be so excessive that the advancement of the piston is not able to compress the steam fast enough to keep up the pressure, the compression line will fall away as at *e*.

The Compression Line.—When there is no expansion, the steam required to fill the clearance space is a dead waste. With a cut off engine it gets a chance to expand with the other steam, and does some good; but still there is a saving by compression, and theoretically by compression up to the initial pressure.

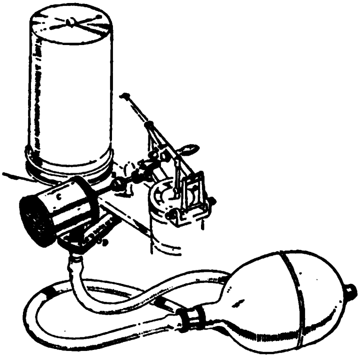


FIG. 2,576.—Robertson & Sons pneumatic pencil control attachment. By squeezing the ball, the pencil is brought in contact with the paper on the drum.

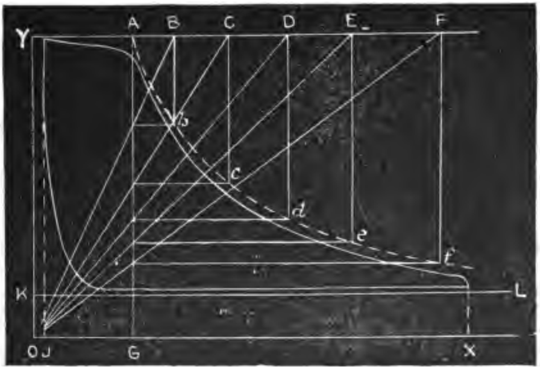


FIG. 2,577.—The hyperbolic expansion curve. The expansion line on an indicator card from the point of cut off to pre-release should be an adiabatic curve, that is a curve which represents neither gain nor loss in heat. As the curves on actual cards never follow the line it is a good plan to draw an adiabatic curve on the indicator card for reference. To draw this curve accurately requires the use of a rather complicated formula, but it very nearly corresponds to an hyperbola, and the latter can be drawn by a simple graphical method. In the diagram, the dotted line represents the curve so drawn. The vacuum line OX, is drawn at a distance to scale, below the atmospheric line KL, corresponding to barometer pressure, practically 14.7 lbs. OY, is the clearance line. YF, is drawn parallel to OX. through the point of highest steam pressure shown on the card. Continue the expansion line up until it meets this line at A. Select any number of points B, C, D, E, F. Connect each of these points to O, and draw the line AG, parallel to OY. From the points where the radial lines cross the line AG, draw horizontal lines and from the points B, C, D, E, F drop vertical lines. Through the points where these two sets of line meet, draw the curve *bcdef*, which is an exact hyperbola. As a matter of fact the true adiabatic curve is slightly below the curve as drawn above, but this curve is accurate enough for all practical purposes. The saturation curve or the relations between pressures and volumes as given in steam tables is also sometimes drawn on an indicator card. Generally this curve will fall between the hyperbola as drawn above and the real expansion line of the indicator card.

When the engine is of a type in which the compression is constant, as in four valve engines, the best results will be attained under normal loads by having the compression round up nicely into the admission line, as at *a*, fig. 2,571, meeting the perpendicular line at about one-third of its height. This will require a different setting of the exhaust valve for different heights of the counter-pressure line.

Figs. 2,572 to 2,575 are examples of compression line corresponding to various conditions as described under the figures.

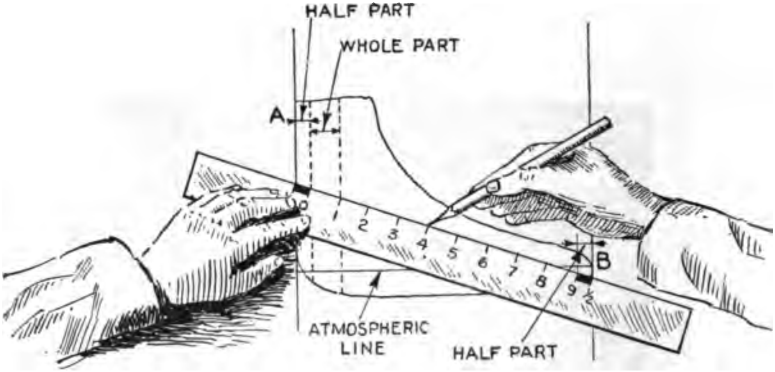


Fig. 2,578.—Method of finding the mean effective pressure from the diagram by averaging its ordinates. Divide the length of the card as indicated into a number of equal parts, say 10, setting off half a part at **A**; half a part at **B**, and nine other parts thus found. Erect ordinates cutting the diagram; and perpendicular to the atmospheric line at the points of division, add together the length of these ordinates intercepted between the upper and lower lines of the diagram and divide by their number. This gives the mean height which, multiplied by the scale of the indicator spring, gives the mean effective pressure (*m.e.p.*).

The Atmospheric Line.—This line is drawn after taking the diagram, and shutting off the steam connection between the indicator and the engine cylinder. The atmospheric pressure is acting on both sides of the indicator piston, allowing the spring to expand into its original shape, thus indicating atmospheric pressure. In closing the indicator cock, care should be taken to close it so that the small drip hole drilled into the plug at right angles with the steam passage opens communication between the atmosphere and the indicator, otherwise an error may occur.

The Vacuum or Zero Pressure Line.—The line of perfect vacuum is a line drawn below the atmospheric line at a distance measured on the scale of the spring used in taking the diagram, equal to 14.72 lbs. referred to a 30 inch vacuum.

The Clearance Line.—The percentage of clearance of an engine can be found by setting the engine on the center, and filling the clearance space with water, which must be carefully measured.

The position of the cross head must now be marked on the guide, and the engine moved off the center.

Next pour an equal amount of water into the cylinder, on the same end as it took to fill the clearance space, and move the engine toward the same center as before, until the water is just beginning to run over, where it has been poured in.

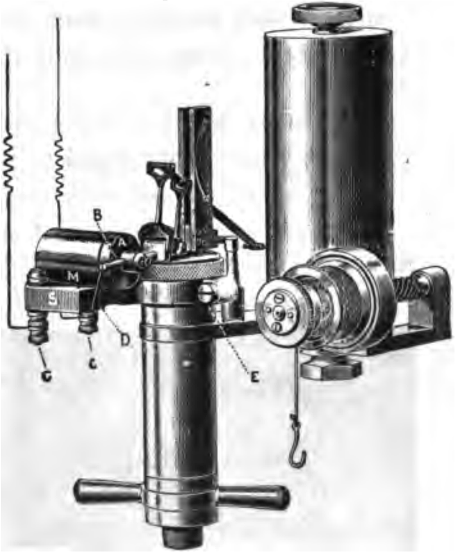


FIG. 2,579.—Electric pencil control as applied to the Taber indicator. ***It consists of*** a magnet M, carried on the support S, which is clamped to the body of the indicator by the screw E. C and D, are the terminals of the magnet windings. An armature D, is mounted on the rod B. The rod B, is screwed into the upright on the swivel plate, and any movement of the armature A, produces a similar movement of the pencil to or from the drum. Spring D, opposes the magnetism and holds the armature and pencil in off position when there is no current through magnet.

Now mark the position of the cross head again, and measure the distance between the marks.

The ratio of this distance to the whole stroke is the ratio of the clearance volume to the whole cylinder volume. The percentage of clearance can also be found from the indicator diagram as in fig. 2,581.

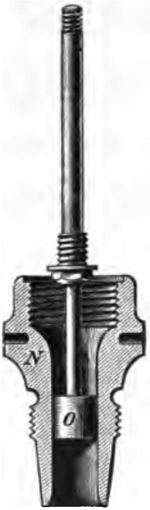


FIG. 2,580—Specially small piston and cylinder end for indicator, shown in fig. 2,508. The lettering is referred to in that figure.

To Find the Mean Effective Pressure from the Diagram.—The mean effective pressure is the mean pressure in pounds per square inch acting on the piston. It is equal to

mean forward pressure — mean back pressure

and as obtained from the indicator card it is equal to

mean height × scale of spring

the mean height being equal to

area of card ÷ its length

Fig. 2,578 shows the method of finding the mean effective pressure from the diagram by averaging its ordinates.

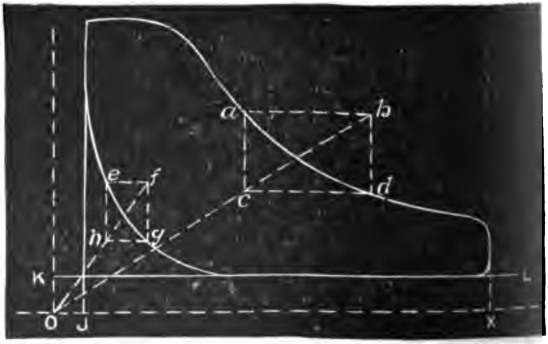


FIG. 2,581.—Method of determining clearance for the diagram. Draw the vacuum line OX, parallel to the atmospheric line KL. Take two points as *a* and *d*, on the expansion line or *e* and *g*, on the compression line, and construct rectangles through these points. Through rectangles thus constructed, draw a diagonal, and at the point O, where it intersects the vacuum line, draw the clearance line OK, and a parallel line tangent to the diagram. The ratio of clearance then is OJ ÷ JX. ***In this method*** of determining clearance, a number of diagrams should be laid out as above and the average taken. The diagonals through the expansion and compression rectangles ***will usually not intersect on the vacuum line:*** in such case clearance line should be drawn midway between the two intersections.

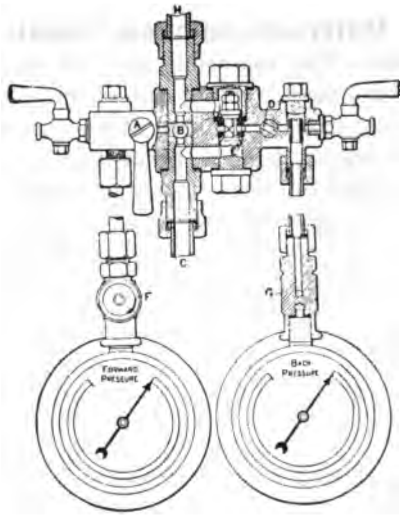


FIG. 2,582.—Ripper mean pressure indicator. Its object is to obtain from pressure gauges a continuous reading of the mean effective pressure in an engine cylinder. The instrument consists of a valve box containing two valves, and by the automatic action of the valves, the driving or impelling steam is made to act continuously on one gauge. called the forward pressure gauge; while the back pressure steam acts continuously upon another gauge, called the back pressure gauge. The difference between the readings of the two gauges gives, for ordinary cases, a close approximation to the effective pressure acting on the piston as given by an ordinary indicator. The action of the valves is as follows: One of the valves, B, is a ball valve, and the other, E, a double seated valve. Suppose, in a vertical engine, the driving steam is on the upper side of the engine piston, pressing it downward, then the driving steam enters also the upper part of the instrument at H, and presses down both the little valves upon their respective seats. This action puts the driving steam into communication with the forward pressure gauge; and puts the back pressure steam, which is below the valves, into communication with the back pressure gauge, owing to the double beat valve E, being now open at the bottom side of the valve. On the return stroke of the piston, the driving steam enters the instrument at C, and the valves B and E, of the instrument are automatically reversed, and again the driving pressure steam acts upon the forward pressure gauge, and the back pressure steam upon the back pressure gauge. In this way there is a continuous reading of the forward and back pressures on the respective gauges. There is a cock A, and also D, at the instrument end of the gauge syphon for rough adjustment, and cocks F and G, close to the gauge for fine adjustment. By the use of two cocks, the gauge finger is maintained steady, and the gauge pipe is kept full of water. The mean pressure obtained from the gauges is the mean pressure on a time base. This differs somewhat from the mean pressure on a distance base, as given by the ordinary indicator, because the motion of the piston is harmonic, and not uniform throughout the stroke. In many cases the difference between the two kinds of mean pressures is very small. In some cases, however, where there is a large expansion in one cylinder, the difference is greater; also in the case where the back pressure at compression is greater than the forward pressure; the valves of the instrument reverse too early. But for all cases, in any given engine, there is a definite ratio between the reading of the gauges and the mean pressure by an ordinary indicator which can be determined once for all by actual trial, or by measurement from the diagrams; and in practice the gauges are in the first instance standardized against a good standard indicator of the ordinary type, at light medium and heavy loads, and the gauges graduated accordingly.

Theoretical Water-consumption Calculated from the Indicator Card.—The following method is given by Prof. Carpenter (*Power*, Sept., 1893): p = mean effective pressure, l = length of stroke in feet, a = area of piston in square inches, $a \div 144$ = area in square feet, c = percentage of clearance to the stroke, b = percentage of stroke at point where water rate is to be computed, n = number of strokes per minute, $60\,n$ = number per hour, w = weight of a cubic foot of steam having a pressure as shown by the diagram corresponding to that at the point where water rate is required, w' = that corresponding to pressure at end of compression.

$$\text{No. of cubic feet per stroke} = l\left(\frac{b+c}{100}\right)\frac{a}{144}.$$

$$\text{Corresponding weight of steam per stroke in lbs.} = l\left(\frac{b+c}{100}\right)\frac{a}{144}w$$

$$\text{Volume of clearance} = \frac{lca}{14{,}400}.$$

$$\text{Weight of steam clearance} = \frac{lcaw'}{14{,}400}.$$

$$\left.\begin{array}{l}\text{Total weight of}\\ \text{steam per stroke}\end{array}\right\} = l\left(\frac{b+c}{100}\right)\frac{wa}{144} - \frac{lcaw'}{14{,}400} = \frac{la}{14{,}400}[(b+c)\,w - cw'].$$

$$\text{Total weight of steam from diagram per hour} = \frac{60\,nla}{14{,}400}[(b+c)\,w - cw'].$$

The indicated horse power is $plan \div 33{,}000$. Hence the steam consumption per hour per indicated horse power is

$$\frac{\frac{60\,nla}{14{,}400}[(b+c)w - cw']}{plan \div 33{,}000} = \frac{137.50}{p}[(b+c)\,w - cw'].$$

Changing the formula to a rule, it is expressed as follows:

Rule.—*To the percentage of the entire stroke which has been completed by the piston at the point under consideration add the percentage of clearance. Multiply this result by the weight of a cubic foot of steam, having a pressure of that at the required point. Subtract from this the product of percentage of clearance. multiplied by weight of a cubic foot of steam having a pressure equal to that at the end of the compression. Multiply this result by* 137.50 *divided by the mean effective pressure.**

The beneficial effect of compression in reducing the water consumption of an engine is clearly shown by the formula. If the compression be carried to such a point that it produces a pressure equal to that at the point under consideration, the weight of steam per cubic foot is equal, and $w = w'$. In this case the effect of clearance entirely disappears, and the formula becomes $137.5\ (bw) \div p$.

In case of no compression, w' becomes zero, and water rate

$$=137.5[(b+c)w] \div p.$$

Prof. R. C. Carpenter (*Sibley Jour. of Eng.*, Dec., 1910) states that tests of engines show that economy is really decreased by high compression. Armand Duchesne (*Power*, Jan. 10, 1911) gives as a reason for this that the steam undergoing compression is superheated and the work of compressing the superheated steam is greater than the work which it gives out later *when it is in the condition of saturated steam.*

The author believes that any decrease in economy claimed to be due to high compression, is not occasioned by the work of superheating the steam but, 1, by decreased mechanical efficiency and 2, by leakage past piston and valve during compression. Considering the amount of water in the cylinder at the beginning of compression, due to condensation, it is doubtful in most cases, if the steam ever reaches the superheated state during compression.

Prof. Denton (*Trans. A. S. M. E.*, xiv, 1363) gives the following table of theoretical water consumption for a perfect Marriott expansion with steam at 150 lbs., above atmosphere, and 2 lbs. absolute back pressure:

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*NOTE.—For compound or triple expansion engines read: *divided by the equivalent mean effective pressure, on the supposition that all work is done in one cylinder.*

*NOTE.—This method applies only to points in the expansion curve or between cut off and release.

Theoretical Water Consumption

Ratio of expansion	M.E.P., lbs. per sq. in.	Lbs. of water per hour per horse power, W.
10	52.4	9.68
15	38.7	8.74
20	30.9	8.20
25	25.9	7.84
30	22.2	7.63
35	19.5	7.45

The difference between the theoretical water consumption found by the formula and the actual consumption as found by test represents "water not accounted for by the indicator," due to cylinder condensation, leakage through ports, radiation, etc.

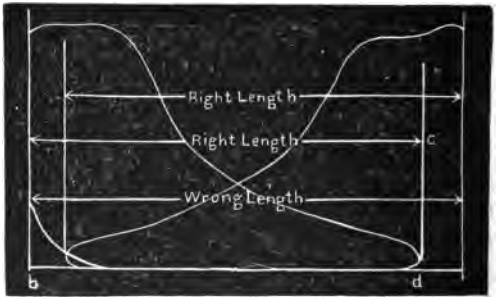


FIG. 2,583.—Diagram illustrating correct length of card. In finding the mean effective pressure from the indicator diagram by *dividing the area by length of the diagram*, care should be taken to measure the correct length. If both diagrams are taken on one card, do not measure between the extremes of both diagrams at once, but measure each diagram by itself as shown in the illustration.

The author believes that any theoretical calculation of water consumption, except to compare the theoretical and actual performance, is a waste of time as it gives no indication of the amount of water used by the engine. There are too many factors, that cannot be calculated, such as initial condensation, re-evaporation, leakage, etc. A much better method of "guessing at" the water consumption is to assume the water consumption to be the same as that obtained by tests of some other engine of the same type and size running under the same, or approximately the same conditions of initial pressure, cut off and back pressure. Obviously, only by tests, or reference to tests can any reliable value be obtained.

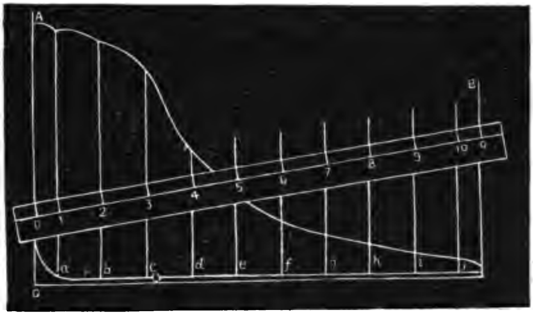


FIG. 2,584.—Ordinate scale for measuring the diagram. Several "short cuts" may be resorted to for shortening the labor of dividing the diagram and locating the ordinates. The simplest of these is to have a rule, a little longer than the ordinary length of diagrams, divided as here shown, just as the diagram is to be divided, with nine spaces of equal length in the middle; the two end spaces, 0 to 1, and 10 to 0, being one-half the width of the others. Four inches between the zero marks of the rule is a good length for diagrams from 3½ to 4 inches in length. Draw the lines O A and X B, at the extreme ends of the diagram and perpendicular to the atmospheric line. Place the rule between them, at such an inclination that both zeros come upon the perpendiculars. Then with a needle-point prick the card opposite each division of the rule, and draw the ordinates perpenducular to the atmospheric line and through these points. Of course any other number of ordinates than 10 or 20 may be used.

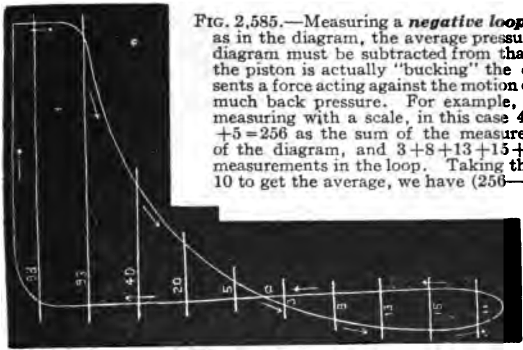


FIG. 2,585.—Measuring a ***negative loop***. If there be a negative loop as in the diagram, the average pressure of the loop portion of the diagram must be subtracted from that of the other portion. For the piston is actually "bucking" the engine, and the loop represents a force acting against the motion of the engine equivalent to so much back pressure. For example, erecting the ordinates and measuring with a scale, in this case 40, we have 98+93+40+20+5=256 as the sum of the measurements in the main portion of the diagram, and 3+8+13+15+11=50 as the sum of the measurements in the loop. Taking the difference and dividing by 10 to get the average, we have (256—50) ÷ 10=20.6 lbs. ***m.e.p.***

NOTE.—Strip of paper method of measuring the diagram; also method of measuring *negative loop*. Instead of measuring each ordinate with the scale corresponding to the spring with which the diagram was taken, some engineers prefer to lay off the lengths of the ordinates continuously on the edge of a strip of paper, then to measure the whole length with a scale of common inches, and multiply the length by the scale of the spring. The result will be the height of a rectangle, whose base is equal to the length of the diagram, and whose area equals the area of the diagram. If there be a negative loop in the diagram, as here shown, the average pressure of the loop portion of the diagram must be subtracted from that of the other portion, as the loop represents *negative work*. For example, erecting the ordinates as before directed, and measuring with a scale, in this case 40, we have 98+93+40+20+5=256 as the sum of the measurements in the main portion of the diagram, and 3+8+13+15+11=50 as the sum of the measurements in the loop. Taking the difference and dividing by 10 to get the average, we have $\frac{(256-50)}{10}=20.6$ lbs. ***m.e.p.***

The student of steam engine economy will gain much valuable information by a study of Geo. H. Barrns' called Engine Tests, also the various tests of Profs. Jacobus, Denton, and others.

In forming an opinion from a test or set of tests as to the value of any feature of operation, as for instance, superheating, jacketting, condensing, etc., *it is only by the closest scrutiny of every condition entering the test and a considerable knowledge of steam engine performance that a misleading conclusion can be avoided.* Thus the steam jacket has been condemned by some and thought to be of very little value by others. As a matter of fact, the trouble is not with the jacket but due to bad judgment in using it under unsuitable conditions.

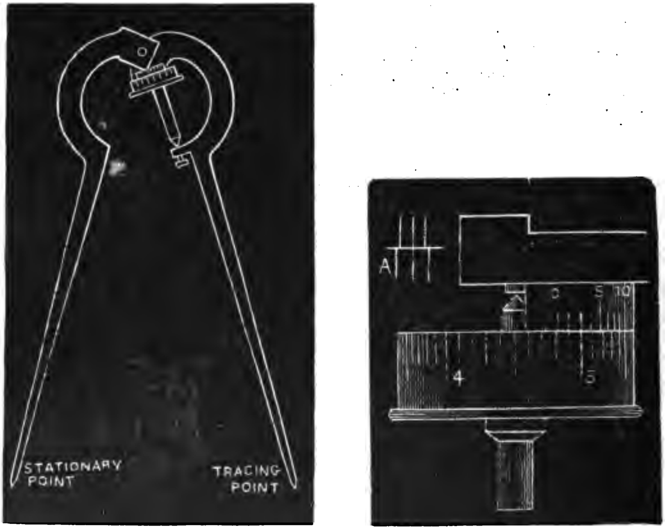


FIG. 2,586 and 2,587.—Amsler planimeter and detail of vernier. ***It consists of*** two arms pivoted together at the top, upon one of which is carried a roller free to revolve upon an axis parallel to the arm itself. The roller is divided circumferentially into ten equal parts, each of which represents a square inch of area, and each of these parts is further divided into equal parts representing each one-tenth of a square inch, as shown in fig. 2,588. Close to the edge of the roller is a stationary plate having the same curvature and containing a vernier made by dividing a space nine-tenths as long as one of the large divisions of the roller into ten equal parts.

Tests have shown that the use of jackets have given a range of results from actual loss to a saving of 40 per cent. The latter saving having been obtained by Donkin with a small engine at an early cut off.

It must be obvious that the benefit of the jacket increases with the number of expansions, and little consideration will show that results obtained by tests at various cut offs give only apparent values, which do not represent the full gain, for the loss due to clearance increases as the cut off is shortened. The reason for this is because the clearance in per cent referred to the point of cut off increases as the cut off is shortened.

The Planimeter.—For ordinary purposes, the ordinary method of finding the area of indicator cards is sufficiently accurate, but if great accuracy be required a planimeter should be used.

By its use the area of the diagram is measured, and by simply dividing this area by the length of the diagram, the average

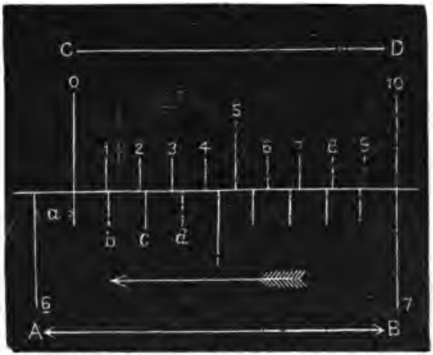


FIG. 2,588.—Vernier scale. In the figure let the space between A and B, represent one of the larger divisions of the planimeter wheel, and the space between C and D, the vernier. In reading the instrument, take the number on the wheel that has passed the zero mark of the vernier when the wheel is turning to the left as indicated by the arrow, as the number of whole square inches, in this case 6. The tenths of a square inch are indicated by the number of spaces, such as *a*, which have passed the zero mark, in this case 1; so that the reading of the scale is 6.1, square inches. Now notice that the space between the line *b*, and the line 1, on the vernier is just one-tenth of one of the spaces such as *a*, upon the roller, the space between the lines 2 and *c*, is just two-tenths, between 3, and *d*, three-tenths, etc. If, then, the wheel roll in the direction of the arrow one-tenth of one of the spaces *a*, corresponding to an area of one one-hundredth of a square inch, the lines 1 and *b*, will coincide, for two one-hundredths, 2 and *c*, would coincide, so that it gives the hundredths of a square inch by writing that number on the vernier which is opposite any line on the wheel. For instance, in reading the instrument as it stands in fig. 2,587, write first the number on the wheel to the left of the zero mark, in this case 4; then the number of whole spaces between that number and the zero mark, in this case 7; and last the number on the vernier which is in line with a mark on the wheel, in this case 3. The whole reading therefore is 4.73 square inches, the decimal point being placed after the 4, the 7 and 3, being tenths and hundredths as before explained. It will be noticed that only the zero, 5, and 10, are numbered on the vernier in fig. 2,587, and this is the case in the actual instrument, and the intermediate marks being easily known by their position. The eye soon becomes accustomed to quickly determine the mark upon the vernier which coincides with one upon the wheel, the marks at either side of it being just within the marks upon the wheel, giving the arrangement shown at A, in fig. 2,587.

height is found. If the area of a rectangle and its length be given, the height can easily be found by dividing the area by the given length.

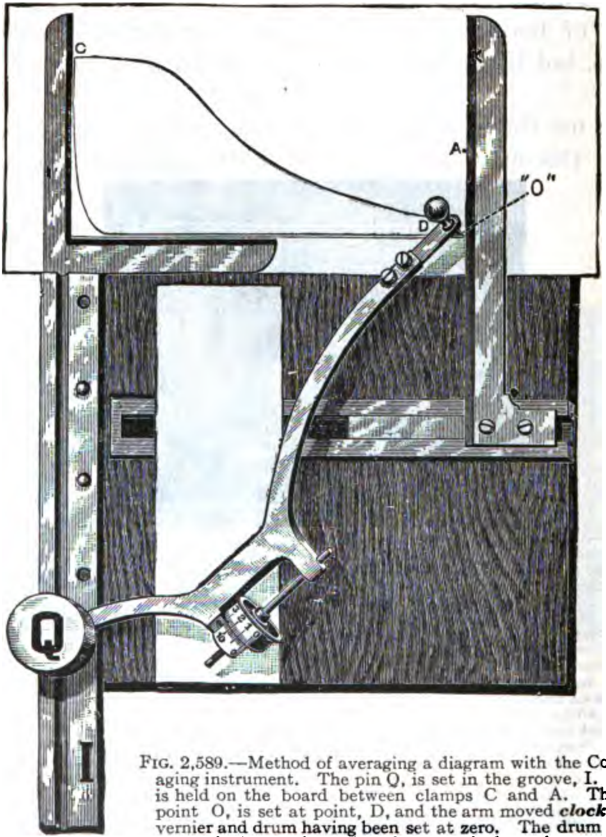
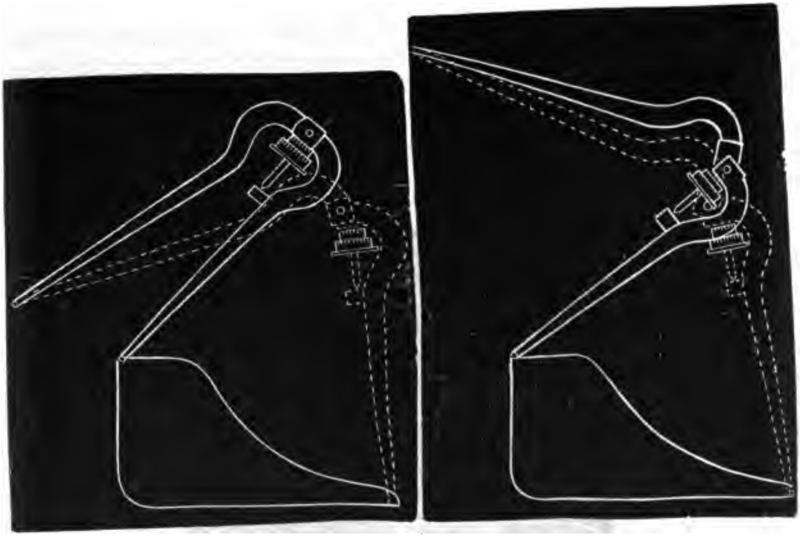
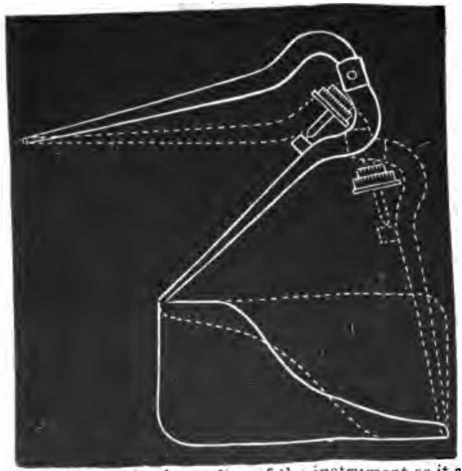


FIG. 2,589.—Method of averaging a diagram with the Coffin averaging instrument. The pin Q, is set in the groove, I. The card is held on the board between clamps C and A. The tracing point O, is set at point, D, and the arm moved ***clockwise,*** the vernier and drum having been set at zero. The drum measures square inches and tenths of square inches, the vernier hundredths. If, after the tracing point has circumscribed the diagram, the drum shows 2 inches and 6-tenths, and the vernier 4 beyond the last graduation on the drum, the area of the diagram will be 2.64 inches. This result is then divided by the length of the diagram, to give the average height, and the quotient multiplied by the number of the indicator spring used in making the card, in order to find the ***mean effective pressure.***





FIGS. 2,590 to 2,592.—Method of using the planimeter. The instrument can be worked to the best advantage when it is neither allowed to close up too closely, as in fig. 2,590, nor to extend too, widely, as in fig. 2,591. A better position for the stationary point than either of these is shown in fig. 2,592, the motion of the roller being easiest when the arms are near a rectangular position. Where the areas to be measured are large, or where there is considerable space between the top of the daigram and the top edge of the card, contact of the roller with the edges of the card, may be avoided by inverting the diagram, as indicated by the dotted diagram in fig. 2,592, using the planimeter always in the same direction—that in which the hands of a watch run; for obviously the area of the diagram remains the same in whatever position the card is placed. Place the tracing point on any convenient point in the line of the diagram, and *press upon it, get a slight indentation to mark the point of starting.* Take the reading of the instrument as it stands; then with the tracing point follow the line of the diagram in the direction in which the hands of a watch move. If the pointer trace in the opposite direction to the hands of a watch the wheels will take out the area instead of adding it.

Now, if the area of an indicator diagram be given, and divided by its length, the result will be the height of a rectangle, of equal area and length with the diagram.

This would be the kind of figure marked upon the paper by the indicator, pencil, if a pressure equal to the mean average should act upon the piston throughout the entire stroke.

The planimeter is made in a variety of forms. The Amsler was the first to be introduced, a typical example of which is shown in fig. 2,586.

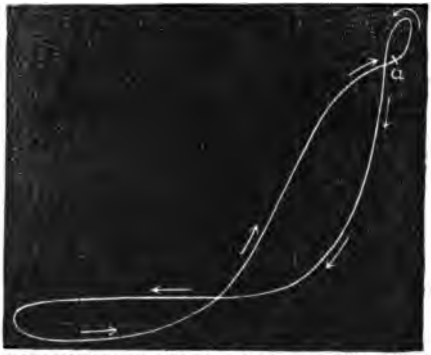
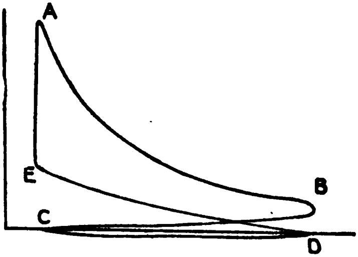


FIG. 2,593.—Tracing a looped diagram with the planimeter. Place the tracing point on any convenient point in the line of the diagram, *and by pressing upon it, get a slight indentation to mark the point of starting.* Take the reading of the instrument as it stands, then with the tracing point follow the line of the diagram *in the direction in which the hands of a clock move.* Remembering that a *clockwise movement of the tracing point adds the area and a counter clockwise movement subtracts,* follow the diagram naturally all the way around as indicated by the arrows in the cut. Do not leave the admission line at *a*, and run out on the back pressure line, but follow the diagram naturally all the way around, as the arrows indicate. If the pointer trace counter clockwise, the wheel will subtract the area instead of adding it. Since the areas of the loops represent *negative work*, it must be subtracted from the other portion of the diagram to get the effective area. Hence, by following the lines of the diagram as directed, the point will pass around the negative portions or "loops" of the diagram counter clockwise, and automatically subtract these areas. Care should be taken in starting, to move the tracing point in such direction that it will move clockwise over the main or positive portion of the diagram.

The planimeter should be used upon a smooth but not slippery surface, such as that of heavy drawing paper or Bristol board. Place a sheet of this large enough to include the planimeter and the diagram upon the drawing board, which is furnished with many planimeters made of hard wood polished, and fasten it with thumb tacks or paper clips, which resemble those on the paper drum of an indicator, and are fastened to the board furnished with the planimeter.

Set the stationary point of the planimeter into the paper in such a position that the tracing point can be carried around the outline of the diagram without bringing the wheel in contact with the card.

Indicating Gas Engines.—While the indicator is sometimes used for determining the horse power of gas engines, especially those of large size, it can be made to serve a number of other useful purposes, as to show the timing, over richness or leanness of mixtures for different conditions of atmospheric pressure and humidity, the proper timing of the exhaust and inlet valve events, whether the exhaust and inlet valve passages are of the right proportion or have too many bends, etc.



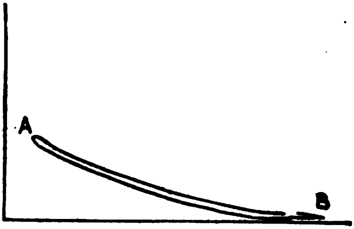
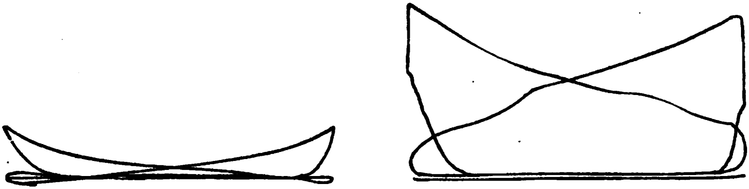


FIG. 2,594.—Typical gas engine card taken with a heavy spring, except that the suction and exhaust lines have been exaggerated so as to be visible. The line CD, represents the suction stroke. If this line drop much below the atmosphere line it indicates that the area of the suction pipe is too small, that the valve opening is too small, or it may be caused by too many bends or short elbows in the suction pipe. DE, represents the compression line, at E, the explosion occurs, and the pressure jumps quickly up to the point A. Line AB, represents the expansion of the gases. At B, the exhaust valve opens. This should be at from 85 to 90 per cent of the stroke. BC, represents exhaust of the pushing out of the burned gases. The exhaust line should be close to the atmospheric line. If the line BC, be too high, it may indicate that the exhaust line is choked, or it may also be due to the cause mentioned above concerning the suction line. It should be remembered that the area included between the suction and exhaust line represents negative work, therefore this area should be kept as small as possible. The above card is for a four-cycle engine.

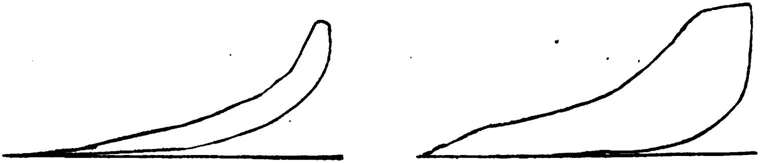
FIG. 2,595.—Card indicating idle stroke of a hit or miss governed engine. The two curved lines represent compression and expansion. If there be no leaks and the cylinder be not cooled too much, these lines should very nearly coincide. Any area between them represents lost work.

For use with gas engines, the regular piston is replaced by one of smaller size, inserting it in a sleeve in the main cylinder. This is done to reduce the area of piston because the pressures in gas engine cylinders are considerably greater than in steam engines.

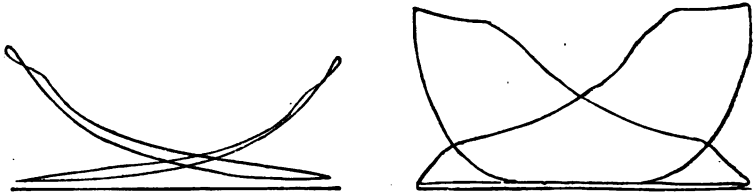
Characteristic Diagrams.—If an engine be in good condition, so that there is no leakage in the valve or piston, and everything well adjusted, it should give a good card.



FIGS. 2,596 and 2,597.—Single valve throttling engine. These diagrams were taken with the same spring ad illustrate very well the action of the governor. Fig. 2,596 shows a light load and fig. 2,597 a heavy load. The point of cut off always remains at the same point, about three-quarters stroke, but as the load increases, more steam is admitted to the cylinder. and the initial pressure is consequently higher.



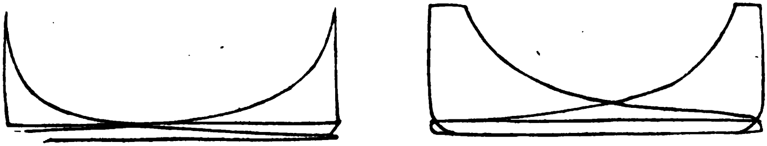
FIGS. 2,598 and 2,599.—Single valve cut off engine. For light and heavy loads, the initial steam pressure is about the same, and as the load increases, the point of cut off occurs later in the stroke. Compression becomes later as the cut off increases. The points of admission and release change also, but not so noticeably as the cut off and compression.



FIGS. 2,600 and 2,601.—High speed (250 *r.p.m.*) single valve automatic engine. The first diagram shows the "friction load" and indicates the work done in overcoming friction in the bearings of the engine. The diagrams for heavier loads usually show large compression at these speeds, and the point of cut off is seldom sharp.

In practice all kinds of distortions are produced due to leakage, faulty setting of the valves, and in some cases, bad design or errors in construction of the valve gear. Figs. 2,596 to 2,605 show diagrams typical of various kinds of engine, and figs. 2,606 to 2,619 various faults which are frequently found in engine operation and which are, as can be seen, readily detected by the indicator.

Indicating Compound and Triple Expansion Engines. —Indicator diagram multi-stage expansion engines are taken in the same manner as are simple engines. To obtain accurate results a separate indicator should be used on each cylinder and all cards taken at the same time.



FIGS. 2,602 and 2,603.—Moderate speed Corliss engine. Fig. 2,602 represents a load lighter than economical and fig. 2,603 about ordinary load. When the valves are in good order there is always a characteristic sharp cut off. Small compression is required because of slow speed and quick admission. The exhaust line being below the atmospheric, indicates that the engine is running condensing. The amount of vacuum in pounds on the return stroke is the distance between the exhaust and atmospheric lines measured on a scale corresponding to the spring used.

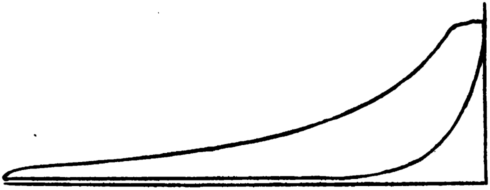


FIG. 2,604.—Four valve engine of the new poppet valve type. This is a characteristic diagram of one of the newest types of Nordberg Poppet valve engines non-condensing speed 200 *r.p.m.* *M.e.p.* 31.75.

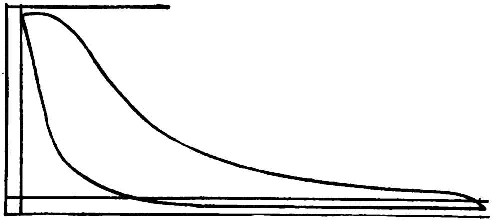


FIG. 2,605.—Uniflow engine. This is the characteristic card of one of the latest types of Nordberg Uniflow Engines, operating under normal load, condensing.

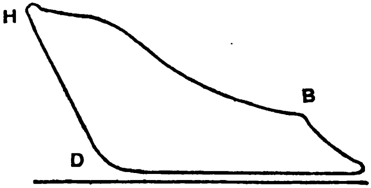
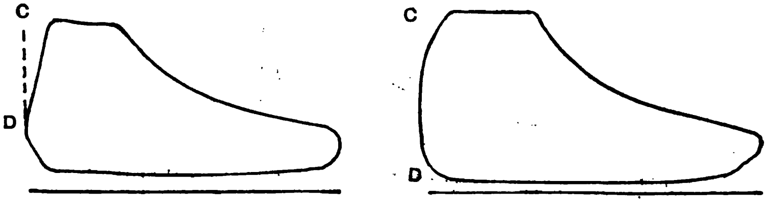


FIG. 2,606.—*Admission too early.* The line HD, slants backward instead of being nearly perpendicular. In a single valve engine, the cut off, release and compression also will be too early as shown, but in a Corliss these parts may not be affected. In the single valve engine, the eccentric should be shifted on the shaft to decrease the angular advance. In the Corliss the lap of the admission valve should be increased. When release occurs too early as shown at B, there is a loss of pressure due to the escape of steam. The valve should be set so that the drop from the expansion line to the back pressure line is as near as possible to the end of the stroke.



FIGS. 2,607 and 2,608.—*Admission too late.* The admission line CD, slants forward instead of being nearly perpendicular. With the Corliss or releasing gear it results from valves becoming misplaced or badly set. Admission may be made earlier by reducing the lap of admission valves. With single slide valve engines all other events are late also, as shown in the second diagram. The release occurs so late there is excessive back pressure at the beginning of the return stroke. The valve should be set so that the drop from expansion line back to pressure line is at the end of the stroke.

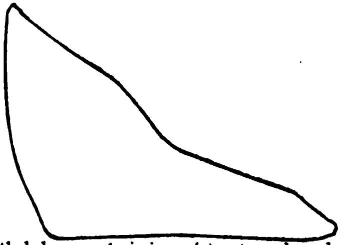
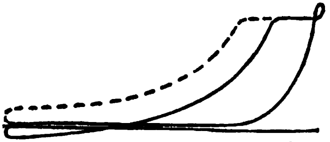


FIG. 2,609.—Steam throttled during admission. The steam line drops instead of being horizontal, indicating that the port opening is not adequate to supply sufficient steam, resulting in a loss of pressure commonly called *wire drawing.*



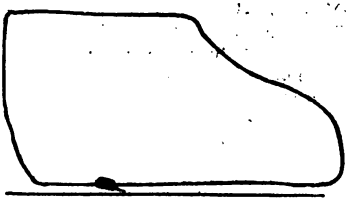
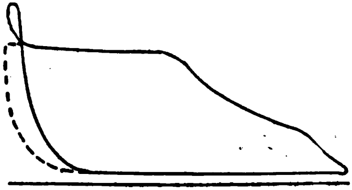


Fig. 2,610.—***Cut off too early.*** Steam expands until its pressure is less than the back pressure, and the expansion line crosses the back pressure line forming a loop. This is frequently found in automatic cut off engines working on light load. On such engines it is caused by too high steam pressure and wastes steam for the amount of work done, besides causing reversal of pressure and pounding. It cannot be remedied by re-setting the valve. Steam must be throttled, the back pressure lowered by running condensing, or the engine run slower. The dotted line shows the result of slowing down the engine and cutting off later. This diagram shows also the result of too early compression. The steam is compressed in the clearance space until the pressure rises above that in the steam chest.

Fig. 2,611.—***Cut off too late.*** This is characterized by a high pressure at release and means a big waste of steam due to its inability to do more work by expansion before the exhaust opens. The boiler pressure should be raised so that a diagram of equal area may be obtained with an earlier cut off, or the revolutions of the engine increased, and the cut off made earlier.



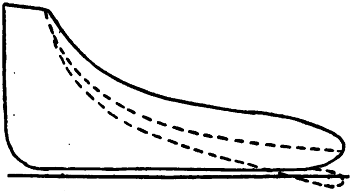
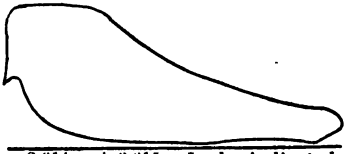
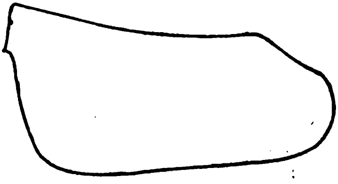


Fig. 2,612.—***Compression too early.*** The pressure in cylinder rises above that in the steam chest, and may cause chattering of the valve, pounding and heating, in addition to reducing the effective work of the steam. The dotted area plus the loop indicate the additional work obtained by setting the valves for proper compression. A similar loop may be caused by the absence of load on the steam valve, but when this fault occurs, the loop comes to a sharp point at the top.

Fig. 2,613.—***Leaks indicated by the expansion line.*** If the expansion line be noticeably above the theoretical adiabatic expansion line, steam enters the cylinder through a leaky valve after cut off. If the expansion line fall below the theoretical, steam leaks past the piston, or gets out through a leaky exhaust valve.





Figs. 2,614 and 2,615.—***Leaks indicated by compression line.*** The curvature of the compression line changes as the piston nears the end of the stroke, sometimes even forming a hook. This indicates the escape of steam from the compression space, either through the exhaust or past the piston.

Figs. 2,617 to 2,620 show the three cards taken from a triple expansion engine. The cards for the different cylinders were taken with different springs in order that the areas of the cards might be large enough to show up well, and when so combined their expansion lines should closely follow the hyperbolic curve representing the total range of expansion.

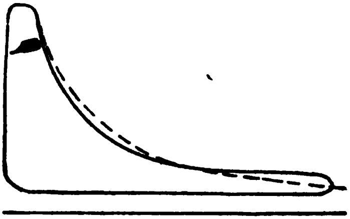
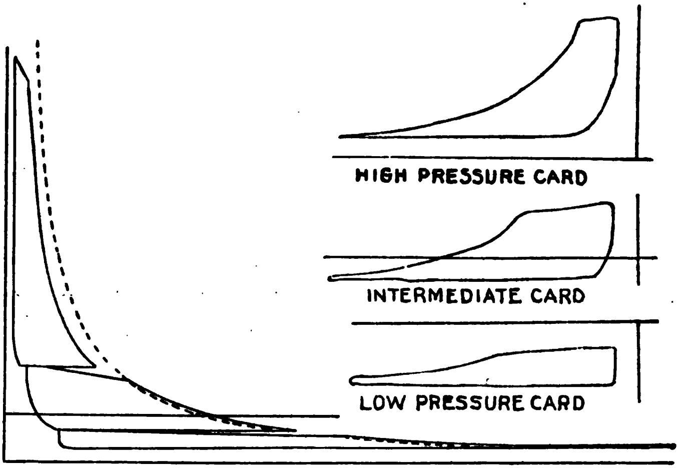
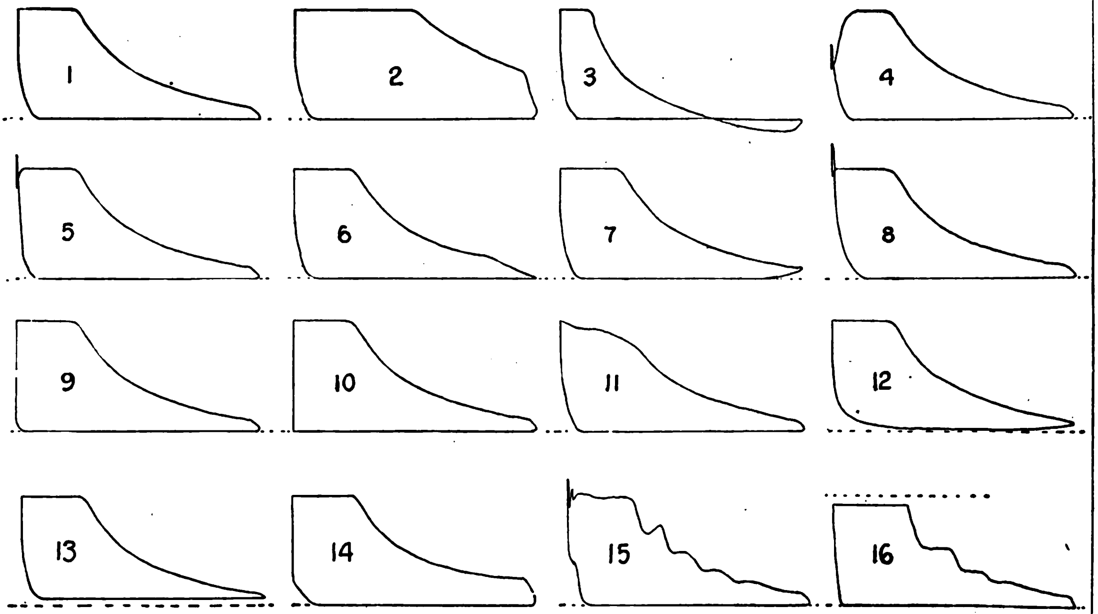


Fig. 2,616.—***Condensation and re-evaporation in cylinder.*** If the expansion curve fall below the theoretical adiabatic line at the beginning of the stroke, crosses, and then goes above, it is a fair sign that excessive condensation occurs at the beginning of the stroke, and re-evaporation during the latter part. Superheated steam or a jacketted cylinder will remedy such trouble, and restore the expansion line to approximately the normal position.



Figs. 2,617 to 2,620.—Indicator cards from triple expansion engine and combined diagrams. From the cards, figs. 2,618 to 2,620, it is seen that the high pressure cylinder is working entirely above atmospheric pressure; the intermediate, both above and below, while the low pressure cylinder works entirely below atmospheric pressure. In order that these cards may appear in a proper relation to each other, they can be combined in one diagram as in fig. 2,617. *To do this the pressure and volumes in the several cylinders have to be reduced to a common scale,* as explained in a previous chapter, and when so laid out the different cards fall in the positions as shown. ***In the combined diagram,*** the total length of each card represents the volume of the corresponding cylinder.



FIGS. 2,621 to 2,636.—Various indicator cards illustrating good and faulty operation. No. 1, excellent steam distribution. No. 2, cut off too late. No. 3, cut off too early. No. 4, late admission or no lead. No. 5, early admission or too much lead. No. 6, pre-release, too early. No. 7, pre-release too late. No. 8, compression too early. No. 9, compression too late. No. 10, no compression. No. 11, choked admission. No. 12, choked exhaust. No. 13, excessive back pressure. No. 14, leaky valves. No. 15, inertia of indicator. No. 16, sticky indicator.

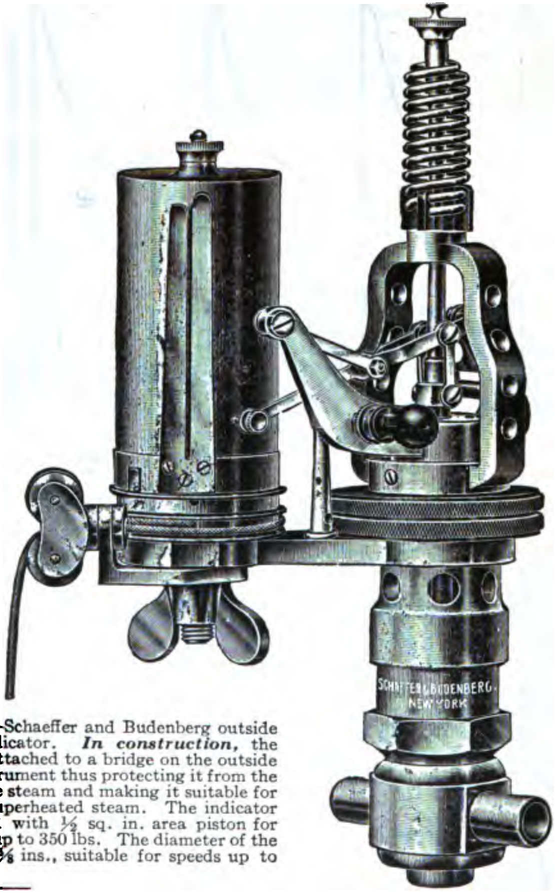


FIG. 2,637.—Schaeffer and Budenberg outside spring indicator. *In construction,* the spring is attached to a bridge on the outside of the instrument thus protecting it from the heat of the steam and making it suitable for use with superheated steam. The indicator is supplied with ½ sq. in. area piston for pressures up to 350 lbs. The diameter of the drum is 1⅝ ins., suitable for speeds up to 600 *r.p.m.*

NOTE.—***The continuous card indicator*** is an instrument so constructed that a number of consecutive diagrams can be taken giving comprehensive and more complete data of the action of the engine under varying loads and during extended tests. In place of the ordinary drum, the continuous drum, containing a small roll of paper, is fitted. With the return stroke the paper is automatically shifted, thus from fifty to a hundred diagrams may be taken on one strip of paper. The atmospheric line is also automatically drawn by means of a point fitted for that purpose.

NOTE.—Schaeffer and Budenberg gas engine indicators are supplied with three pistons and cylinders have ½, ¼ and ⅛ sq. in. areas, adapting the instrument to low, medium and heavy pressures respectively. A safety stop is provided for protecting the springs when subjected to pressures beyond their range limit.

READY REFERENCE INDEX

A

Adhesive power, locomotives, increasing, 957.
Adjustable spanner wrench, ills., 1,177.
Admission line, ills., 1,285, 1,287, 1,289, 1,292, 1,310.
Air brake, locomotive, *air compressor*, 1,032–1,044.
    governor, ills., 1,026.
  air strainer, Westinghouse, ills., 1,028.
  automatic, ills., 997, 998.
  valve, automatic, ills., 1,007, 1,009, 1,010, 1,012.
    check, ills., 1,028.
    distributing, 1,003, 1,006, 1,008, 1,011, 1,017.
    feed, 1,023, 1,024.
    independent, 1,018–1,023.
    positions, loco., 1,014, 1,015, 1,016 1,017, 1,020, 1,031.
    quick acting, 1,005, 1,017.
    safety, E-6, ills., 1,005.
    triple, 997.
    Westinghouse, ills., 999–1,029.
Air compressor, locomotive, 1,032–1,044.
  governor, ills., 1 026, 1,027.
  pump, direct connected, ills., 1,145–1,147, 1,162.
  valve, combination, ills., 1,028
Alignment, engine, 1,100–1,114.
Arrester, spark, locomotive, 967.
Ash pan, locomotive, 958.
Assembling steam engine, 1,108.
Atmospheric pressure; indicator, ills., 1,234.
Automatic, *air brake*, equipment, 1,040–1,043.
    valve, ills., 1,007–1,009.
  control, locomotive, diag., 1,030.
  engine, induction diagram, 1,308.
  grease cup, ills., 1,140.
  valve, accumulator accessories, 1,248.
    relief, accumulator accessories, 1,248.
Axle, locomotive, 953.

B

Babbitting bearings, method, 1,183.
Baffle, plate, locomotive, 967.

Baffle,—*Continued*
  turbine, bearing, ills., 1,251.
Baldwin locomotive, American type, ills., 956.
  Atlantic type, ills., 963.
  consolidation type, ills., 959.
  decapod type, ills., 960.
  eight coupled type, ills., 959.
  Forney type, ills., 968.
  four coupled two rear wheel truck type, ills., 967
  Mallet articulated type, ills., 962.
  mogul type, ills., 958.
  Sante Fe type, ills., 960, 977.
  six coupled switching, ills., 969.
  ten wheel type, ills., 956.
  two cylinder compound valves, diag., 989.
  variable cut off, ills., 953.
Bar, boiler, locomotive, ills., 973.
  guide, locomotive, 982.
  radius, locomotive, ills., 964.
Beam engine, see walking beam engine.
Bearing(s), babbiting, method, 1,183.
  gland, turbine, ills., 1,219.
  hot, causes, 1,157.
  indicator, lubrication, 1,277.
  locomotive running gear, 991.
  lubrication, 1,123.
  out board, aligning, ills., 1,103.
  reduction gear, turbine, ills., 1,267.
  scraping, ills., 1,113.
  self-aligning, ills., 1,158.
  self-oiling, 1,134, 1,158.
  shaft, turbine, ills., 1,219.
  step, turbine, ills.. 1,251.
  thrust, generator, ills., 1,244.
    turbine, 1,222.
  turbine, ills., 1,219, 1,222, 1,251.
Bed plate alignment, ills., 1,112.
Bell crank, indicator, ills., 1,282.
Belpaire boiler, locomotive, des., 970.
Belt, clamp, home made, ills., 1,156.
  dressing, application, diag., 1,155.
  drive, countershaft location; power station, ills., 1,099.
  lacing, diag., 1,157.
  power, 1,153.
  running, 1,154.
Box, journal, locomotive, ills., 996.
  sand, locomotive, 955.
Boxwood indicator scale, ills., [illegible]

Brake, locomotive, see Air brake.
location, 955.
Bituminous coal grate, ills., 972.
Bianca's, steam turbine, ills., 1,207.
Blake, jet condenser, ills., 1,146, 1,148.
Black lead, 1,119.
Blast pipe, locomotive, 963.
use, 963.
Blower, locomotive, 976.
Boiler, connection to header, ills., 1,101.
dry pipe, ills., 1,095.
erecting, 1,104.
header, connecting, ills., 1,101.
horizontal, dry pipe, ills., 1,095.
tubular, supporting, ills., 1,104.
location, in power house, 1,092.
locomotive, 951.

vertical, ills., 1,073.
multi-tubular, 952.
power station, location, ills., 1,092.
pressure, decrease, 1,101.
marine engine, 1,049.
room, plan, 1,092, 1,093.

stop valve, ills., 1,101.
Bolts, anchor, ills., 1,097, 1,098, 1,107.
engine foundation, ills., 1,097, 1,098, 1,099, 1,107.
Broad gauge track, standard, 957.
Buckets, turbine, 1,214.
Buffalo "Spiro" steam turbine, ills., 1,262.
Bumper, locomotive, ills., 995.
Burning point, lubricant, 1,118.
Bushings, indicator, 1,277.
By-pass valve, locomotive, ills., 988.
turbine, ills., 1,235.
Curtis, ills., 1,255.

C

Cab, locomotive, 920.
interior, ills., 977.
Cam drive, poppet valve engine, 1,061, 1,062.
Canvas packings, des., 1,172.
Capillary lubrication system, 1,132, 1,134.
Carbon packing, turbine, ills-, 1,221.
Card, indicator, see Indicator card.
Cast nozzle, steam turbine, ills., 1,203.
Castings, steam engine, 1,097.
Cement, engine foundation, 1,095.
Center, line, wire, securing, ills., 1,102.
pin, locomotive, 991.
Centering piston, steam engine, 1,179, 1,180.
Centrifugal, force, turbine, 1,203.
lubrication, 1,131, 1,135, 1,136.
shaft governor, turbine, ills., 1,234.
Chain, oiling, ills., 1,135.
wind, diag., 1,190.
Chambers, locomotive, 1,036.
Check valve, *locomotive*, 962, 1,001.
air brake, ills., 1,028.

Check valve,—*Continued*
lubricator, ills., 1,130.
steam accumulator, 1,248.
Chesapeake & Ohio Mikado type locomotive, ills., 987.
Choke, locomotive, ills., 1,028.
Circulating pump, 1,145, 1,146.
Clamp, belt, ills., 1,156.
Clearance, turbine, determining, 1,256.
line indicator card, 1,295, 1,296.
Clermont, Fulton's, 1,045.
Clipper engine, marine, 1,064.
Clutch, Hill, ills., 1,168, 1,169.
pulley, friction, Hill, ills., 1,166.
Coal, capacity, locomotive tender, 996.
transportation, ills., 1,092.
Cock, two-way indicator, ills., 1,281.
Coffin averaging instrument, ills., 1,304.
Cold test, lubricant, ills., 1,117, 1,118.
Compound, *engine*, indicator, diagram, 1,309.
marine, 1,046, 1,070–1,073.
two cylinder, crank sequence, 1,075.
impulse turbine, diag., 1,210, 1,212.
locomotive, 988.
marine engine, ills., 1,046, 1,070–1,073.
Graham, for steamer *Stornoway I*, ill., 1,072.
Marine Iron Works, steepie, ills., 1,073.
turbine, 1,213.
multi-stage, 1,213, 1,220, 1,221, 1,250.
reaction, 1,227.
valves, locomotive, 989.
Compression, *line*, indicator card, ills., 1,285, 1,291, 1,292.
indicator, ills., 1,285, 1,286, 1,292.
system, lubrication, 1,,131-1139.
Compressor, governor, locomotive air brake, SF-4, ills., 1,026, 1,027.
locomotive air brake, 1,033-1,044.
Concrete, engine foundation, 1,095-1,100.
Condensation, steam engine, excessive, 1,144.
Condenser, paddle engine, ills., 1,057.
flooding, 1,145.
indicator, 1,288.
induction, Corliss engine, ills., 1,150, 1,151, 1,164.
jet, starting with, ills., 1,146–1,148, 1,163.
lubricator, ills., 1,126, 1,127.
siphon, 1,147, 1,149, 1,163.
surface, operation, 1,146.
Condensing marine engine, 1,046.
tube, lubricator, ills., 1,125.
Connecting, arm, indicator, 1,272.
rods, locomotive, 953, 983.
Consolidation type locomotive, Baldwin, ills., 959.
Control, locomotive, diag., 1,030.
turbine throttle valve, ills., 1,242.
Cord indicator, ills., 1,276, 1,280, 1,283, 1,284, 1,286, 1,300.
pulley, ills., 1,286.
replacement, ills., 1,283.
take up, Trill, ills., 1,280.
whipping, diag., 1,279.

- Corliss engine, condenser, connection, ills., 1,149, 1,150.
- Corrugated furnace boiler, locomotive, 971.
- Counterfeit marine engines, 1,067.
- Countershaft, ills., 1,100.
  - location, power station, ills., 1,099.
- Coupling, power transmission, cut off, ills., 1,171.
  - rods, locomotive, 963, 983.
  - steam turbine, ills., 1,244.
- Cow catcher, locomotive, 955.
- Crank(s), oiler ring, ills., 1,136.
  - *pin*, alignment, ills., 1,111.
    - oil cup, ills., 1,136.
    - oiling, 1,135.
    - repairing, diag., 1,188.
  - sequence of, compound engines, diag., 1,075.
  - web, repairing, diag., 1,188.
- Cross head, *drive*, pump, marine eng., 1,085.
    - indicator, ills., 1,276, 1,282.
  - guides, steam engine, testing, 1,109.
  - locomotive, type, 976, 980–982.
  - lubrication, 1,138.
- "Crown" sight feed oil cup, ills., 1,131.
- Curtis turbine, ills., 1,238, 1,250.
  - clearance, ills., 1,256.
  - construction details, ills., 1,256, 1,257, 1,259.
  - governor, ills., 1,231, 1,234.
  - lubricaton system,
  - nozzle, ills., 1,203.
  - packing ring, ills., 1,255.
  - starting, 1,253.
  - step bearing, ills., 1,251.
  - stuffing box, ills., 1,255.
  - *valve*, ills., 1,255.
    - gear, ills., 1,254.
- Curve, expansion, hyperbolic, diag., 1,293.
  - railroad, standard, 957.
- Cut off, coupling, power transmission, ills., 1,171.
  - early, ills., 1,311.
  - engine, indicator card, 1,308.
  - gear, Sickles, 1,049.
  - late, ills., 1,311.
  - valve, ills., 1,061, 1,063.
  - variable, locomotive, ills., 953.
  - walking beam engine, 1,049.
- Cylinder, air compressor, 1,034.
  - cap, distributing valve, ills., 1,017.
  - *engine*, alignment, 1,100, 1,103, 1,108, 1,109.
    - arrangement, marine engine, 1,077.
    - assembly, 980, 1,098, 1,099.
  - blowing out, 1,110.
  - condensation, ills., 1,312.
  - Corliss, ills., 1,146.
  - head, tightening, 1,185.
  - lubrication, 1,124, 1,125, 1,128, 1,130.
  - *marine engine*, ills., 1,050, 1,077, 1,078, 1,081, 1,085.
    - warming, 1,145.
  - locomotive, 953, 978, 980.

D

- Dampers, locomotive, 966.
- Deflectors, smoke box locomotive, 974.
- Decapod type locomotive, ills., 960.
- DeLaval turbine, bearings, ills., 1,222.
  - buckets, fastening, ills., 1,214.
  - governor, ills., 1,228.
  - multi-stage, ills., 1,213, 1,215.
  - operation, 1,266–1,268.
  - packing, 1,221.
  - reduction gear, ills., 1,266, 1,267.
  - simple impulse, ills., 1,265.
  - valve, throttle, ills., 1,230.
  - wheels, attaching, ills., 1,217.
- Detroit lubricator, ills., 1,127.
- Diagram, factor, triple engine, 1,081.
  - indicator, see Indicator card.
- Direct connected, engine, ills., 1,056, 1,058.
  - pump, ills., 1,145, 1,162.
- Disassembling steam engine, 1,176.
- Doctor feed pump, western river, 1,064.
- Draught, locomotive, forced, 963.
- Drivers, locomotive, 957, 987.
- Driving, axle, locomotive, ills., 985.
- Drum, indicator, 1,269, 1,270, 1,280.
- Dry pipe, boiler, ills., 1,095.
  - locomotive, 973.
- Duplex gauge, air brake, ills., 1,004.
  - indicator spring, ills., 1,273.

E

- Eccentric, backing, marine engine, 1,049.
  - locomotive, ills., 984, 985.
  - lubrication, 1,138.
  - marine engine, 1,049.
  - placing, valve setting, 1,056.
  - *rod*, beam engine, ills., 1,050, 1,051.
    - paddle engine, ills., 1,056.
- Ejector, marine engine, 1,161.
- Engine, see Steam engine.
  - locomotive, see Locomotive.
  - lubrication, 1,115–1,122.
  - marine, 1,045–1,090.
  - room, plan, 1,092, 1,093.
  - rotary, 1,193–1,200.
  - steam, see Steam engine.
- Engineer's packing tools, ills., 1,170.
- Exhaust, nozzle, locomotive, 963, 988.
  - *pipe*(*s*), indicator, 1,271, 1,292.
    - locomotive, 963.
  - port, locomotive, 998.
  - steam passages, locomotive, 955.
  - *valve*, indicator, ills., 1,285, 1,286, 1,294.
    - locomotive, 1,004.
    - setting, indicator, 1,294.
    - wipers, ills., 1,050.
- Expansion, curve, hyperbolic, diag., 1,293.
  - packing, diag., 1,164.
  - quintuple, marine engine, author's indicator diags., 1,090.

Expansion,—*Continued*
  - valve, steam accumulator, 1,248.

- Expected diagram, triple expansion marine engine, 1,081.

F

- Feed, oil cup, lubrication, ills., 1,138.
  - *pump*, boiler, care of, 1,160.
    - marine engine, conn., ills., 1,083.
    - starting, 1,146.
    - steam engine, 1,159.
- Fire, box, locomotive, ills., 953, 965, 970, 971.
- Flange, locomotive, 962.
- Fly wheel, engine ass., 1,098, 1,108, 1,109.
- Force feed, lubricatiors, 1,129.
- Forced draught, locomotive, 963.
- Fore and aft compound marine engine, of author's steamer *Stornoway I*, ills., 1,072.
- Formula, piston diameter, locomotive, 981.
- Forney type locomotive, Baldwin, ills., 968.
- Foundation engine, 1,095–1,100.
- Friction, cause of, 1,116.
- Fulton's steamboat Clermont, 1,045.
- Furnace, locomotive, 952.

G

- Gas engine, indicator, diagram, ills., 1,307.
- Gasket, 1,166.
- Gauge, air brake, ills., 1,004.
  - broad, standard, 957.
  - cocks, locomotive, 976.
  - narrow, locomotive, ills., 959.
  - water, locomotive, 976.
- Gear, bearings, ills., 1,266.
  - *locomotive*, 952, 953.
    - running parts, 990, 991.
  - *reduction*, turbine, ills., 1,212, 1,266.
    - locomotive, 953, 980, 990, 991.
- Governor, compressor, locomotive air brake, ills., 1,026, 1,027.
  - indicator, ills., 1,308.
  - intermittent admission, 1,233.
  - throttling, 1,227.
  - turbine, ills., 1,227–1,235, 1,241, 1,253, 1,264.
- Graham, *compound*, engine of steamer *Stornaway I*, ills., 1,072.
  - single acting, one valve triple expansion marine engine, ills., 1,078.
  - special two cylinder, transfer expansion oscillating marine engine, ills., 1,074.
- Graphite, forms, 1,119.
  - packings, applying methods, des., 1,172.
- Grate, area, locomotive, 971.
  - boiler, locomotive, ills., 972, 973.
  - locomotive, 959, 972.
- Grease cups, 1,139–1,141.
- Grouting space, engine bed, ills., 1,099.
- Guide, engine, alignment, ills., 1,106.

H

- Hamilton-Holzworth turbine, 1,218, 1,219.
  - governor, ills., 1,233.
- Hammer, water, steam engine, 1,144.
- Handle, automatic brake valve, ills., 1,020.
- Heater, water locomotive, 962.
- Hedley's locomotive, ills., 952.
- Hemp packings, 1,172.
- Hero's turbine, ills., 1,207.
- High speed marine engine, the author's single acting, one valve triple expansion, ills., 1,078.
  - the author's special two cylinder oscillating transfer expansion, ills., 1,074.
- High vacuum, turbine, 1,249.
- Hill clutch mechanism, ills., 1,166, 1,168.
- Hit or miss gas engine, diag., ills., 1,307.
- Horizontal, adjustment, steam engine shaft, ills., 1,105.
  - boiler, supporting, ills., 1,104.
  - engine, marine, 1,046.
- Hot bearings, steam engine, cause of, 1,157.
- Hot well, steamer *Stornoway I*, ills., 1,160.
- Houghtaling reducing motion, ills., 1,278.
- Hult rotary engine, ills., 1,197, 1,199.
- Hydrokinetic lubricators, 1,123, 1,124, 1,126, 1,128.
- Hyperbolic expansion curve, 1,243.

I

- Independent, brake valve, parts, 1,018–1,022.
  - pump, ills., 1,145, 1,146, 1,149, 1,159.
- Indicator, steam, 1,269–1,314.
  - atmospheric line, 1,294.
  - averaging instrument, ills., 1,304.
  - calculations, 1,296–1,313.
  - cock, ills., 1,294.
  - cord, ills., 1,280, 1,283, 1,284, 1,286, 1,297.
  - cylinder, 1,269, 1,271, 1,296.
  - drum, 1,269, 1,280.
  - inertia of, indicated, 1,313.
  - inside spring type, ills., 1,270, 1,271.
  - lever reducing motion, ills., 1,276.
  - mean pressure, ills., 1,297.
  - outfit, ills., 1,284.
  - outside spring type, ills., 1,280.
  - parallel motion, ills., 1,272.
  - parts, ills., 1,270, 1,280.
  - pencil control, ills., 1,293, 1,295.
  - piping, 1,279, 1,281.
  - piston, 1,269, 1,271, 1,276, 1,296.
  - planimeter, ills., 1,302, 1,305.
  - reducing motion, ills., 1,276, 1,279.
  - scale, ills., 1,274, 1,279.

Indicator,—*Continued*
- speed, Starrett's, ills., 1,159.
- *spring*, 1,273, 1,274, 1,275.
  - changing, ills., 1,275.
  - indicated, 1,313.
- vernier scale, ills., 1,303.

Indicator card(s), 1,307, 1,308, 1,309, 1,310, 1,311.
- *admission*, choked, 1,313.
  - early, 1,310.
  - late, 1,310.
  - line, 1,284, 1,285, 1,287, 1,289.
  - throttled, 1,310.
- atmospheric line, 1,294.
- back pressure, excessive, 1,313.
- clearance, ills., 1,295, 1,296.
- *compression*, early, 1,311.
  - late, 1,311.
  - *line*, ills., 1,285, 1,291, 1,292.
    - leaks indicated, 1,311.
    - object, 1,291.
- condensation in cylinder, 1,312.
- Corliss engine, 1,309.
- *cut off*, early, 1,311, 1,313.
  - engine, 1,308.
  - late, 1,311, 1,313.
- cylinder, condensation and re-evaporation, 1,316.
- *exhaust*, choked, 1,313.
  - line, ills., 1,285, 1,291.
- *expansion line*, ills., 1,285, 1,289.
  - leaks indicated, 1,311.
- gas engine, ills., 1,307.
- *high*, pressure, 1,312.
  - speed engine, 1,308.
- inertia of indicator, 1,313.
- intermediate, 1,312.
- lead, 1,313.
- leakage, 1,311, 1,313.
- low pressure, 1,312.
- multi-stage expansion engine illustrating diagram factor, 1,082.
- negative loop, 1,301.
- oil engine, ills., 1,469.
- poppet valve engine, 1,309.
- *pre-release*, early, 1,313.
  - line, 1,285, 1,290, 1,291.
- *pressure*, back, 1,313.
  - mean effective, 1,296.
- quintuple expansion marine engine, Graham diagrams, 1,090.
- re-evaporation in cylinder, 1,312.
- scale, 1,301.
- single valve throttling engine, ills., 1,308.
- *steam*, distribution, 1,313.
  - line, ills., 1,285, 1,288, 1,289.
- steamer ***Monmouth***, 1,084.
- sticky indicator, 1,313.
- taking, 1,282.
- theoretical, ills., 1,285.
- throttling engine, 1,308.
- *triple expansion engine*, 1,312.
  - graphical method of proportioning cylinder diameters, 1,080.
- uniflow engine, 1,309.
- vacuum line, 1,294.

Indicator card(s),—*Continued*
- valves leaking, 1,313.
- various, ills., 1,313.
- water consumption, 1,298, 1,300.
- zero pressure line, 1,294.

Injectors, locomotive, 967.

Iron rust cement packings, des., 1,171.

J

Jahn's governor, turbine, ills., 1,229.

Jet condenser, 1,146–1,148, 1,163.

K

Kinetic energy, 1,202.

Knocks, steam engine, 1,155.

Koerting condenser, ills., 1,150.

L

Lacing belt, 1,157.

Laying up steam engine, 1,188, 1,189.

Lead, black, uses, 1,119.

Leaks, indicated, ills., 1,311, 1,313.

Lighter, steam, power plants, author's comparison, diags., 1,068.

Line, see Indicator card.

Link motion, so called Stephenson, ills., 984.

Lippincott indicator, ills., 1,271.

Locomotive(s), 951–1,044.
- adhesive power, 957.
- *air brake*, 1,032–1,044.
  - valve, see Air brake valve.
- air compressor, see 1032–1044.
- American type, Baldwin, ills., 956.
- Atlantic type, Baldwin, ills., 961, 963.
- baffle plate, use, 967.
- bearings, main, 991.
- blast pipe, use, 963.
- *boiler*, attachments, 976.
  - bar, ills., 973.
  - Belpaire, des., 970.
  - classification, 968.
  - fire box, des., 970.
  - frame, 953.
  - grate, variations, 972.
  - heating surface, 971.
  - requirements, 962.
  - standard form, 964.
  - straight top, 969.
  - turtle back, 969.
  - wagon top, 969.
  - water grate, ills., 973.
  - Wooten, 969, 970.
- *brake*, air, 1,032–1,044.
  - location, 955.
  - double heading, 1,023.
- bumpers, ills., 995.
- cab, interior, ills., 977.

Locomotive(s),—*Continued*
- classification, ills., 954–962.
- compound, distribution, diag., 988.
- compressor, 1,032–1,034.
- connecting rod, des., 983.
- consolidation type, Baldwin. ills., 959.
- control, automatic, diag., 1,030.
- coupling rod, des., 983.
- cow catcher, 955.
- cross heads, 981, 982.
- cylinder, casting, parts, ills., 978, 979.
- dead engine, device, 1,028.
- decapod type, Baldwin, ills., 960.
- double heading, 1,023.
- dry pipe, 973.
- eight coupled type, ills., 959.
- exhaust, nozzle, 955, 963.
- fire box, construction, 965, 970, 971.
- forced draught, 963.
- Forney type, ills., 968.
- four coupled, ills., 967.
- freight, cross head, 976, 981.
- fuel, 959.
- gear, running, 990.
- grate area and heating surface, 971.
- guide, 981, 982.
- Hedley's, ills., 952.
- holding, 1,031.
- injectors, use, 976.
- intercepting valve, 987.
- journal boxes, ills., 996.
- Mallet articulated type, ills., 962.
- Mikado type, ills., 961, 987.
- Mogul type, ills., 958.
- "Old Ironsides," ills., 951.
- operation, 1,002, 1,029.
- Pacific type, 961, 964.
- pilot, ills., 995.
- *piston*, 980, 981.
  - rod, 980.
- power, 963.
- Prairie type, 961, 964.
- reducing valve, 989.
- running gear, 990, 991.
- saddle, 953.
- safety valve, "pop," 976.
- sand box, 955, 976, 996.
- Santa Fe type, ills., 960.
- slide valve, 979.
- *smoke*, box, 973, 974.
  - stack, ills., 974.
- spark arrester, 967.
- springs, 991.
- steam dome, parts, 967.
- switching type, ills., 965, 969.
- ten wheel type, ills., 956.
- tender, ills., 994–996, 1,042.
- truck, four wheel, 953, 991–993.
- types, ills., 956, 958–965, 967, 968.
- *valve*, chest, ills., 979.
  - compound, diag., 989.
  - *gear*, ills., 979, 983.
    - link motion, Stephenson, 984.
    - ***Walschaerts***, ills., 984–986.
  - safety, 976.
  - slide, 979.

Locomotive, valve,—*Continued*
  - starting, 988.
  - throttle, 955, 973, 975.
- variable cut off, ills., 953.
- water scoop, ills., 994.
- weight, distribution, 953, 955.

Loose, eccentric reversing gear, 1,049.

Lubricants, 1,115–1,122.
- grease, 1,139.
- oil, 1,119–1,122.
- qualities, 1,116, 1;151.
- temperature effects, 1,117, 1,142.
- tests, 1,116–1,122.

Lubricating, devices, 1,124–1,141.
- *grease*, 1,119, 1,139.
  - cups, 1,140.
- indicators, 1,277.
- *oil*, 1,119–1,122.
  - application, 1,142.
  - cups, 1,131–1,138, 1,142.
  - grooves, 1,142.
  - pump, 1,128–1,130.
  - rancid, 1,122.
  - tests, 1,116–1,122.

Lubrication, 1,123–1,142, 1,151.
- bearings, turbine, 1,244.
- engine, 1,123.
- forced, turbine, ills., 1,252.
- *systems*, capillary, 1,132.
  - centrifugal, 1,135.
  - compression, 1,139.
  - external, 1,130.
  - gravity, 1,132.
  - inertia, 1,135.
  - pressure, 1,138.
  - splash, 1,141.
- *turbine*, bearing, 1,244.
  - forced, ills., 1,252.

Lubricators, gravity, 1,124.
- hydrokinetic, 1,124.

Lunkenheimer oiling devices, 1,131–1,133.

M

Mallet articulated type locomotive, ills., 962.

Management, steam engine, 1,143–1,192.

Manhole, locomotive, 974.

Manzel oil pump, ills., 1,129.

Marine, *boiler*, piping, ills., 2,264.
  - vertical, Marine Iron Works, ills., 1,073.

Marine engine(s), 1,045–1,090.
- air pump, ills., 1,083.
- alignment, 1,111, 1,112.
- *author's*, 3½ and 8 × 6 compound of steamer ***Stornoway I***, ills., 1,072.
  - oscillating compound, ills., 1,074.
  - single acting, triple expan., ills., 1,074.
- beam, ills., 1,048–1,055.
- bed plate, aligning, ills., 1,112.
- classification, 1,045.
- compound, 1,070–1,075, 1,077.

Marine engine(s),—*Continued*
- counterfeit, 1,067.
- cross head, pump, ills., 1,085.
- cut off valve, ills., 1,063.
- cylinders, classification, 1,047.
- design, 1,059, 1,060, 1,064.
- dipper, 1,064.
- doctor pump, 1,064.
- feed pump, connection, ills., 1,083.
- frames, 1,047.
- inclined paddle, des., 1,057.
- loads, characteristic, 1,047.
- lubrication, 1,134.
- *Marine Iron Works*, adjustable cut off, ills., 1,063.
  - single cylinder tug, ills., 1,065, 1,066.
  - steeple compound, ills., 1,073.
  - stern wheel, ills., 1,058.
- multi-stage, diagram, 1,082.
- nigger, 1,064.
- oscillating, two cylinder transfer expansion Graham special, ills., 1,074.
- paddle, ills., 1,056, 1,057.
- poppet valve, diag., 1,062.
- power application, 1,046.
- *pressure*, 1,049.
  - classification, 1,045.
- propeller, ills., 1,048.
- proportions, 1,060, 1,064.
- *pump*, cross head, ills., 1,085.
  - drive, ills., 1,089.
- quadruple expan., types, ills., 1,086, 1,087.
- quintuple expan., Graham diags., 1,090.
- shaft alignment, 1,111.
- simple, 1,065.
- single cylinder, ills., 1,066.
- steam lighter power plants, author's comparison, ills., 1,068.
- Seabury, ills., 1,079.
- single cylinder, ills., 1,066.
- stern wheel, des., 1,057–1,060, 1,063.
- *triple expansion*, ills., 1,075–1,085.
  - Graham single acting, 1,078.
- tug, ills., 1,065.
- *valve*, adjustable cut off, ills., 1,063.
  - cut off, ills., 1,063.
  - gear, Sweeney, des., 1,062.
  - poppet, ills., 1,061, 1,062.
  - setting, 1,053.
- variable cut off valve, Marine Iron Works, ills., 1,061.
- vertical inverted, des., 1,065.
- walking beam, 1,048–1,055.
- Western river, ills., 1,061.

- Marine Iron Works, boiler, ills., 1,073.
  - engines, ills., 1,058, 1,063, 1,065, 1,066, 1,073.
  - valves, ills., 1,061, 1,063.
- Mean effective pressure, diag., 1,294, 1,296.
- Mean pressure, indicator, ills., 1,297.
- Metallic packing, ills., 1,172, 1,173.
- Mikado type locomotive, ills., 961, 987.
- Mineral oil, 1,121.
- Mogul type locomotive, ills., 958.
- Moncky wrench packing, 1,162, 1,175.
- Monmouth, steamer, indicator cards, 1,084.
- Multi-stage, expansion engine, cut off, 1,048.
  - *marine engine*, diag., 1,082.
    - author's oscillating, ills., 1,074.
  - turbines, see Steam turbines.

N

- Napier's experiment, 1,205.
- Nathan oiling device, ills., 1,124, 1,128.
- Nelseco Diesel engine, ills., 1,475.
- New York loco. compressor, ills., 1,032.
  - control equipment, ills., 1,030.
- "Nigger" engine, marine, 1,064.
- Non-condensing marine engine, 1,046.
- Non-expansive rotary engine, ills., 1,196.
- Nozzle, exhaust, locomotive, 955, 963.
  - lubricator, ills., 1,126.
  - turbine, ills., 1,203–1,205, 1,209.
- Nugent oiling devices, ills., 1,152.
- Nut, obstinate, starting, 1,185.

O

- Oil, animal, 1,119.
  - cups, 1,131–1,138.
  - grooves, ills., 1,158.
  - *lubricating*, 1,119–1,122.
    - application, 1,142.
  - mineral, 1,121.
  - packings, des., 1,168.
  - piping, packing for, 1,168.
  - pump, ills., 1,128–1,130.
  - rancid, 1,122.
  - tests, 1,116–1,122.
  - vegetable, 1,120.
- Oiler, crank pin, ills., 1,136.
  - ring, ills., 1,158.
  - sight feed, ills., 1,132, 1,137.
- "Old Ironsides," locomotive, ills., 951.
- Oscillating engine, Graham, two cylinder transfer expansion marine, jacketted, ills., 1,074.
- Outboard bearing, ills., 1,103, 1,104.
- Over travel, steam engine, 1,156.

P

- Pacific type, locomotive, ills., 964.
- Packing, 1,165.
  - air, des., 1,168.
  - ammonia, 1,168.
  - applying, methods, 1,170–1,173.
  - asbestos, des., 1,173.
  - canvas, 1,172.
  - carbon, ills., 1,221.
  - expansion, diag., 1,164.
  - graphite, 1,172.
  - hemp, 1,172.

Packing,—*Continued*
  iron rust cement, des., 1,171.
  metallic, 1,172, 1,173.
  "Moncky" wrench, 1,162.
  movable, 1,169.
  object, 1,165.
  oil, 1,168.
  paper, 1,172.
  rings, air piston, locomotive, 1,038.
  rubber, 1,173.
  stationary, des., 1,168, 1,171.
  steam, des., 1,167.
  tools, ills., 1,170.
  turbine, ills., 1,221.
  water, des., 1,167.
Paddle, engine, marine, 1,046, 1,056, 1,057.
Pantograph reducing motion, ills., 1,278.
Parallel, motion device, indicator, ills., 1,272.
Parson's steam turbine, 1,223, 1,225, 1,232, 1,237–1,242, 1,260, 1,261.
"Penguin," oil cup, ills., 1,138.
"Pilgrim" oil cup, ills., 1,131.
Pilot, locomotive, ills., 995.
Pin, crank, see Crank pin.
  indicator, 1,272.
Pinion, steam turbine, ills., 1,267.
Pioneer steam boats, 1,045.
Pipes, air brake, locomotive, 1,000.
Pipe, arrangement, ills., 1,092, 1,094.
  bends, indicator, ills., 1,281.
  blast, locomotive, 963.
  cocks, indicator, ills., 1,281.
  connections, engine installation, 1,095.
  dry, locomotive, 973.
  exhaust, locomotive, 962.
  feed valve, locomotive, 1,025–1,026.
  installation, valves, 1,094.
  power station, ills., 1,107.
  sand, locomotive, location, 977.
  tongs, ills., 1,187.
  wrenches, various type, ills., 1,186.
Piping, indicator, 1,279, 1,281.
  oil, steam turbine, 1,248.
  steam, condenser, connection, ills., 1,094.
    length, 1,093.
    power plant, ills., 1,092, 1,094.
    vibration prevented, ills., 1,107.
  steamer, 1,088, 1,161.
Piston, *air*, 1,034.
    compressor, ills., 1,036.
    pump, packing, 1,170.
  assembly, 1,110.
  balanced, turbine, 1,225.
  centering, 1,179.
  clearance, 1,178.
  indicator, 1,269, 1,270, 1,279, 1,296.
  locomotive, 980, 981, 988, 1,036.
    diameter formula, 981.
  packing, 1,171.
  rod, adjustment, 1,156.
    alignment test, ills., 1,180, 1,181.
    centering, ills., 1,180.
    locomotive, 978, 980, 988.
    rust, 1,188.
"Planet," oil cup, ills., 1,136.
Planimeter, ills., 1,302–1,306.
Plant(s), power, steam lighter, diag., 1,068.
  steam, installation, 1,091–1,114.
Plumbago, uses, 1,119.
"Polaris," oil cups, ills., 1,136.
Pony truck, locomotive, 967.
"Pop" safety valve, locomotive, 976.
Poppet valve, cam driven, 1,061.
  engine, beam, diag., 1,052.
  gear, beam engine, 1,049, 1,050.
  locomotive, 975, 976.
  marine engine, ills., 1,061, 1,062.
Powell, lubricators, ills., 1,125, 1,127, 1,137.
  oil cups, ills., 1,131, 1,136, 1,138.
Power, control clutch, ills., 1,166.
  plant, steam lighter, author's diags., 1,068.
  stations, 1,091, 1,092, 1,093.
    pipe, supporting, ills., 1,107.
Power stroke, gas engine, ills., 1,326, 1,349.
  oil engine, 1,475.
  transmission, belt, ills., 1,153.
    cut off coupling, ills., 1,171.
    line shafting, 1,171.
    quill drive, ills., 1,167.
Prairie type, locomotive, ills., 964.
Pressure, boiler, marine, 1,048.
  gauge, indicator, ills., 1,297.
  mean effective, 1,294, 1,296.
  steam, critical, 1,205.
    turbine, see Steam turbine.
  system, lubrication, 1,138.
Pump, *air*, direct con., 1,145, 1,147, 1,16
    independent, ills., 1,146.
    jet, condenser, 1,147.
    starting engine with, 1,145.
    stopping engine with, 1,162.
  boiler feed, care of, 1,160.
  circulating, 1,145, 1,146.
  drive, marine engine, ills., 1,089.
  feed, 1,145, 1,146, 1,149, 1,159.
    operation, 1,160.
    starting, 1,146.
  location, power plant, ills., 1,094.
  marine engine, drive, ills., 1,085, 1,089.
    feed connection, ills., 1,083.
  oil, force feed, 1,129.
    hand, ills., 1,128.
  power, ills., 1,129, 1,130, 1,138.
  tank pump, 1,149.
  valve gear,

Q

Quadruple expan. engine, 1,046, 1,086, 1,087.
Quill drive, ills., 1,167.
Quintuple expansion, engine, author's diags., for 1,000 and 1,500 lbs. pres., 1,090.

R

Radial, arm, indicators, 1,272.
Rateau steam accum., 1,241, 1,242, 1,246.

Reciprocating, engine, over travel, 1,156.
Reducing, motion, indicator, 1,276–1,282.
  wheel, indicator, ills., 1,277.
Rees marine engines, 1,048, 1,059, 1,060.
Relief valve, accumulator, 1,248.
  power plant piping, ills., 1,094.
Reservoir, locomotive air brake, 998, 1,003.
Reversing valve gear, loose eccentric, 1,049.
Ridgway turbine, 1,203, 1,263, 1,264.
Ripper mean pressure indicator, ills., 1,297.
Robertson & Sons, pencil control, 1,293.
Rod, coupling locomotive, 963.
  piston, alignment, 1,180, 1,181.
Rotary, *engines*, 1,193–1,200.
    marine, 1,047.
  turbine, def., 1,193.
Rotor, turbine, ills., 1,206, 1,209, 1,211, 1,213, 1,237.
Rubber packing, ills., 1,173.
Running gear, locomotives, 953, 990, 991.
Rust, steam engine, preventing, 1,188, 1,191.

S

Saddle, locomotive, 953.
Safety valve, air brake, ills., 1,005.
  locomotive, 967, 976.
Sand, box, locomotive, use, 955, 976, 996.
  pipes, locomotive, 977.
Santa Fe type locomotive, ills., 960.
Scale, indicator, ills., 1,274, 1,301–1,303.
  ordinate, 1,301.
  vernier, ills., 1,303.
Screw, marine engine, 1,046.
Schaeffer and Budenberg indicator, 1,273, 1,314.
Schooner, "Santa Anna" machinery arrangement, ills., 1,088.
Scoop, water, locomotive, ills., 994.
Scotch yoke pump drive, ills., 1,089.
Screw, drivers, ills., 1,176.
  piston grease cup, "Marine" type, ills., 1,140.
Seabury marine engines, 1,063, 1,077, 1,079.
Self-oiling bearing, ills., 1,158.
Shaft, *steam engine*, journal, 1,108.
    alignment, 1,105, 1,109, 1,111, 1,112.
    thrust, ills., 1,188.
  steam turbine, ills., 1,217, 1,219.
    governor, ills., 1,234.
Shifting link, locomotive, 952.
Sickles cut off gear, 1,049.
Sight feed lubricating device, 1,127, 1,129, 1,131–1,133, 1,137, 1,138.
"Signal" sight feed oil cup, ills., 1,131.
Simple, impulse turbine, ills., 1,265.
Single, acting, one valve triple expansion marine engine, Graham, ills., 1,078.
  cylinder marine engine, ills., 1,066.
Siphon condenser, 1,147, 1,149, 1,163.
Slide, *valve*, locomotive, 979, 988.
    unbalanced, 1,059.
Smoke, box, locomotive, 973, 974.
  stack, locomotive, ills., 952, 974.
Smith type, Hill clutch, ills., 1,169.
Soapstone, 1,119.
Soule rotary engine, parts, ills., 1,196.
Spark arrester, locomotive, 967.
"Spiro" steam turbine, Buffalo, ills., 1,262.
Springs, indicator, ills., 1,273, 1,275, 1,279, 1,280.
  locomotive, 953.
    running gear, 991.
Standard, gauge track, 957.
  locomotive boiler, 965.
Starrett speed indicator, ills., 1,159.
Station, power, 1,091–1,094, 1,099.
  installation, 1,091–1,114.
  management, 1,143–1,193.
Steam, accumulator, 1,241–1,248.
  air compressor, diag., 1,035.
  boat(*s*), pioneer, 1,045.
    stern wheel, 1,060, 1,064.
  boiler, see Boiler.
  chest, indicator, 1,288.
  critical pressure, 1,205.
  distribution, compound loco., diag., 988.
  dome, locomotive, parts, 967.
  economy, 1,092, 1,093.
  energy, card, 1,240.
Steam engine, *alignment*, 1,100–1,115.
  assembling, 1,108.
  auxiliary machinery, laying up, 1,189.
  beam, 1,048–1,057.
  bearings, see Bearings.
  bed plate alignment, 1,112.
  care, 1,165.
  castings, main, placing, 1,097.
  compound, indicating, 1,309.
    Graham special two cylinder, transfer expansion, jacketted, marine oscillating, ills., 1,074.
    3½ and 8×6, fore and aft, of author's steamer ***Stornoway I***, ills., 1,072.
    two cyl., crank sequence, diag., 1,075.
  concrete bed, ills., 1,098.
  condensation, excessive, effects, 1,144.
  condenser, jet, ills., 1,148.
  Corliss, condenser, con., 1,149, 1,150.
  crank and crank pin, see Crank.
  cross head guides, testing, 1,109.
  cut off, see Cut off.
  cylinder, see Cylinder.
  disassembling, 1,176.
  eccentric, see Eccentric.
  exhaust, see Exhaust.
  feed pump, 1,159, 1,083.
    starting with, 1,146.
  foundations, template, ills., 1,095–1,100.
  gasket, 1,166.
  Graham, special two cylinder, transfer expansion, jacketted, marine oscillating, ills., 1,074.
    3½ and 8 × 6, fore and aft, compound, of steamer ***Stornoway I.***, 1,072.

Steam engine, *Graham*,—*Continued*
single acting one valve triple expansion., 1,078.
oscillating marine, tr. expan., 1,074.
guides, aligning, ills., 1,106.
grouting, ills., 1,099, 1,106.
high pressure, ills., 1,045.
high speed, triple expan., Graham, 1,078.
oscillating, Graham, 1,074.
indicator card, see Indicator card.
installation, 1,091–1,114.
jet condenser, connection, ills., 1,146.
knocks, causes, 1,155.
laying up, 1,188.
leaks indicated, 1,311.
locomotive, see Locomotives.
lubricants, see Lubricants.
lubrication, see Lubrication.
lubricators, 1,124, 1,131–1,133.
oiling, see Lubrication.
operation, 1,144–1,164.
knocks, 1,155.
lubrication, 1,123–1,142, 1,151.
starting, 1,144–1,151.
stopping, 1,160–1,164.
management, 1,143–1,192.
marine, see Marine engines, 1,045–1,096.
multi-stage marine, diag., 1,082.
packing, see Packing.
paddle, 1,046, 1,056, 1,057.
piston, see Piston.
poppet valve, see Poppet valve.
quadruple expansion, 1,046, 1,086, 1,087.
quintuple expansion, author's diag. for 1,000 and 1,500 lbs. pressure.
reciprocating parts, assembling, ills., 1,108.
steam energy, 1,240.
over travel, 1,156.
repair, 1,174.
rotary, 1,193–1,200.
rust, preventing, 1,188.
shaft, alignment, ills., 1,105.
measuring, diag., 1,182.
placing, 1,109.
thrust, ills., 1,188.
stern wheel, 1,057.
throttling, indicator diagram, ills., 1,308.
triple expansion, Graham, 1,078, 1,309.
marine, 1,075–1,085.
oscillating, Graham, 1,074.
two cylinder, propeller, ills., 1,048.
walking beam, 1,048–1,057.
water hammer, 1,144.
working principles, diag., 1,195.
Steam, expansion, turbine, 1,240.
kinetic energy, 1,202.
launch "Stornoway I," hot well, ills., 1,160.
Graham componnd engine, for, 1,072.
machinery arrangement, ills., 1,161.
lighter power plants, author's system, comparison, diags., 1,068.
line, indicator card, 1,285, 1,288, 1,289.
packings, 1,167.
pipe, length, 1,093.

Steam, *pipe*,—*Continued*
vibration, preventing, ills., 1,107.
plant, installation, 1,091–1,114.
reciprocating engines, diag., 1,240.
Steam turbine(s), 1,201–1,268.
Allis-Chalmers, 1,225, 1,261.
bearing, collar, ills., 1,243.
spindle, ills., 1,243.
step, ills., 1,251.
thrust, ills., 1,244.
Bianca's, ills., 1,207.
classification, 1,206, 1,207.
compartments per stage, 1,217.
compound, 1,213.
construction details, 1,256.
coupling, flexible, ills., 1,244.
Curtis, 1,253, 1,256.
cylinder, ills., 1,239.
DeLaval, running, 1,266.
development, 1,201.
discs, stationary, ills., 1,219.
economical operation, 1,249.
generator, Allis-Chalmers, ills., 1,244.
gland, ills., 1,219.
governor, action, 1,230.
auxiliary, ills., 1,235.
classification, 1,227.
Curtis, ills., 1,234.
emergency, ills., 1,253.
high pressure, ills., 1,264.
intermittent, ills., 1,232, 1,233.
Jahn's type, ills., 1,229.
operation, 1,228.
Parsons, ills., 1,232, 1,241.
partial admission, 1,230, 1,231.
shaft, centrifugal, Curtis, 1,234.
throttling, ills., 1,227, 1,228.
Hero's, ills., 1,207.
high pressure, 1,236, 1,263.
horizontal, ills., 1,259.
impulse, 1,203, 1,207, 1,222.
compound, 1,212, 1,220.
description, 1,206–1,209.
multi-stage, 1,215–1,222, 1,250.
simple, ills., 1,210, 1,211, 1,265.
multi-stage, 1,213, 1,215, 1,220.
low pressure, 1,236, 1,238, 1,239.
lubrication system, 1,252.
mixed pressure, 1,236, 1,238.
multi-stage, 1,215, 1,220.
bearings, ills., 1,222.
compound, 1,220, 1,222, 1,250.
DeLaval, ills., 1,213, 1,224.
governor, 1,229.
impulse, 1,215–1,222, 1,250.
packing, ills., 1,221.
simple, 1,215–1,220.
nozzles, ills., 1,202–1,206, 1,217.
oil piping, diag., 1,248.
one stage, ills., 1,256.
operation, ills., 1,201, 1,202, 1,253, 1,260, 1,266.
Parsons, 1,223, 1,225, 1,237, 1,241, 1,242, 1,260, 1,261.
piping oil, 1,248.
principles, 1,201.

Steam turbines,—*Continued*
    radial flow, two stage, ills., 1,236.
    reaction, 1,222.
        Allis-Chalmers, Parsons, ills., 1,223.
        compound, 1,227.
        pressure, 1,211.
        principle, 1,207, 1,208.
    reduction gear, ills., 1,266, 1,267.
    reversing chambers, ills., 1,226.
    rotation, cause, 1,203.
    rotors, ills., 1,237.
    running, 1,260.
    safety stop, ills., 1,242.
    shaft, bearing, ills., 1,219.
        wheels, 1,217.
    spindle, bearing, 1,243.
        glands, ills., 1,245.
    "Spiro," Buffalo, ills., 1,262.
    stage, 1,213.
    starting, Curtis, 1,253.
    steam, flow, 1,204.
        pressure, 1,235.
    steaming, 1,248.
    stuffing box, ills., 1,255.
    Terry, 1,226.
    two stage, ills., 1,257.
    vacuum, 1,249.
    valve, by pass, Curtis, ills., 1,255.
        gear, mechanical, Curtis, ills., 1,254.
        operating mech., hydraulic, 1,232.
        stop, ills., 1,253.
        throttle, ills., 1,230, 1,242.
    water purifying system, diag., 1,247.
    Westinghouse-Parsons, ills., 1,260.
Steamer, Monmouth, indicator card, 1,084.
    Stornoway I, hot well, ills., 1,160.
        engine for, 1,072.
        machinery arrangement, 1,161.
Steeple marine engine, ills., 1,073.
Step bearing, turbine, ills., 1,251.
Stephenson, link motion, ills., 984.
Stern wheel, engines, 1,057–1,064.
    steam boats, proportions, 1,060, 1,064.
Stethoscope, ills., 1,156.
Stornoway I, author's 3½ and 8×6 compound engine for, ills., 1,072.
    hot well, ills., 1,160.
    machinery arrangement, ills., 1,161.
Strap wrench, ills., 1,189.
Stuffing box, 1,157, 1,255.
Sturtevant pressure lubrication, ills., 1,139.
Surface condenser, starting with, 1,146.
Sweeney valve gear, 1,062.

T

Table, locomotive classification, 954.
    steam boat proportions, 1,064.
    water consumption, 1,300.
Tabor indicator, parts, ills., 1,273, 1,295.
Tank, locomotive tender, 995.
Tender, locomotive, 953, 994, 995.
Template, engine foundation, 1,096, 1,097.

Template,—*Continued*
    steam engine, cylinder, axis, ills., 1,100.
        foundation, ills., 1,096.
Ten wheeled type locomotive, ills., 956.
Tender, locomotive, ills., 994–996.
Terry steam turbine, operation, 1,226, 1,236.
Tests, oil, des., 1,121, 1,154.
Theoretical, indicator diag., 1,285, 1,298, 1,300.
    water consumption, table, 1,300.
Thompson indicator, 1,272, 1,273, 1,275, 1,284.
Track gauge, ills., 957.
Tram, measuring shaft right, 1,182.
Trammel, ills., 1,107.
Tool(s), kit, ills., 1,191.
    packing, ills., 1,170.
    pipe, tongs, ills., 1,187.
        wrench, ills., 1,189.
    screw driver, ills., 1,176.
    wrenches, ills., 1,174–1,178, 1,186, 1,189.
Transmission, motion, locomotive, 953.
    power, belt, ills., 1,153.
        cut off coupling, ills., 1,171.
        line shafting, 1,171.
        quill drive, ills., 1,167.
Trill indicator, ills., 1,273, 1,276, 1,277, 1,279, 1,280, 1,286.
Triple expansion engine, indicating, 1,309.
    marine, 1,075–1,085.
        author's jacket'd single acting, 1,078.
        cylinder proportions, 1,080, 1,081.
            various pressures, 1,084, 1,085.
        designing, 1,079.
        diagram factor, 1,081, 1,082.
        diag., steamer ***Monmouth***, 1,084.
        power, distribution, 1,083.
        single acting, Graham, 1,078.
        stroke, 1,083.
        types, ills., 1,076, 1,077.
Truck, *locomotive*, 991–993.
        journal boxes, ills., 996.
        four wheel, 953, 957, 992.
        two wheel, 958, 993.
        pony, 958.
Twin screw steamer ***Monmouth***, indicator cards, 1,084.

U

Unbalanced slide valve, 1,059.

V

Valve, action indicator, 1,284.
    admission, locomotive, 979.
    air brake, see Air brake valve.
    angle, ills., 1,101.
    assembly, 1,110, 1,111.
    boiler stop, 1,101.
    bucket, ills., 1,145, 1,146.
    by pass, turbine, ills., 1,235.

- Valve,—*Continued*
  - check, ills., 962, 1,161.
  - chest, ills., 979, 1,050, 1,051.
  - compound locomotive, diag., 989.
  - cut off, see Cut off.
  - diagram, poppet, beam engine, 1,052.
  - discharge pump, 1,094.
  - distributing, 1,000–1,008, 1,011.
  - *exhaust*, lap, 1,157.
    - leak, 1,292.
    - wiper, ills., 1,050.
  - expansion, steam accumulator, 1,248.
  - feed, locomotive, ills., 1,024.
  - foot, ills., 1,145, 1,146, 1,150.
  - *gear*, assembly, 1,110.
    - beam engine, ills., 1,049, 1,050.
    - link motion, ills., 983–987.
    - locomotive, 979–983–987.
    - loose eccentric, 1,049.
    - marine engine, 1,048–1,050, 1,059, 1,062.
    - non-releasing, direct, 1,048.
    - steam turbine, mech., Curtis, 1,254.
    - walking beam engine, 1,049, 1,050.
    - ***Walschaerts***, 984–986.
  - head, ills., 1,145, 1,146.
  - inside admission, ills., 979.
  - indicated, 1,292.
  - *locomotive*, air brake, see Air brake valve.
    - parts, ills., 979.
    - safety, 976.
    - slide, 979.
    - throttle, 973, 975.
  - non-return boiler stop, ills., 1,101.
  - oil, regulating, ills., 1,133.
  - pilot, 1,232.
  - piston, ills., 979, 1,177.
  - *poppet*, beam engine, diag., 1,052.
    - cam driven, 1,061.
    - double seated, ills., 1,050.
  - pump, ills., 1,145.
  - reducing, locomotive, 989, 1,034.
  - *relief*, 1,161.
    - water seal, on exhaust, 1,094.
  - reversing, 1,036.
  - setting, 1,053, 1,185.
  - slide, locomotive, 979.
  - starting, locomotive, ills., 988.
  - stern wheel engine, 1,059.
  - stop, 1,101, 1,161.
  - throttle, 955, 973, 975, 1,230.
  - unbalanced, slide, 1,059.
  - vacuum, 1,229.
  - variable cut off, ills., 1,061.
- Vacuum, jet condenser, 1,147.
  - piping arrangement for, 1,094.
- Variable cut off, locomotive, ills., 953.
  - valve, Marine Iron Works, ills., 1,0
- Vauclain locomotive, steam dist., 988.
- Vernier scale, ills., 1,303.
- Vertical, boiler, marine, ills., 1,073.
  - engines, 1,046, 1,065.

W

- Walking beam engine, 1,048–1,057.
- ***Walschaerts*** valve gear, locomotives, 98 986.
- Water, capacity locomotive tender, 995.
  - consumption, theo., calc., 1,298–1,300.
  - gauge, locomotive, 976.
  - grate, boiler, locomotive, ills., 973.
  - hammer, 1,101, 1,144.
  - purifying system, turbine, 1,247.
  - scoop, locomotive, ills., 994.
  - seal, valve, ills., 1,094.
- Wheels, locomotive, 953, 956, 961, 967.
- Well, hot, of str. ***Stornoway I***, 1,160.
- Western river, marine engine, ills., 1,060 1,061, 1,062, 1,064.
- Westinghouse, *air brake*, equipment, 997–1,028, 1,034–1,043.
  - air compressor, 1,035–1,043.
  - reducing, ills., 1,034.
  - safety, ills., 1,005.
  - *turbine*, ills., 1,260.
    - nozzle chamber, ills., 1,205.
    - reduction gear, ills., 1,212.
    - rotor, ills., 1,209, 1,211.
    - valve, ills., 1,235.
- Wrist pin, locomotive, 980.
  - lubrication, 1,133.
- Whyte's locomotive class., table, 954.
- Wooten boiler, fire box, ills., 970, 971.
- Worthington jet condenser, ills., 1,148.
- Wrenches, ills., 1,174–1,178.

Y

- Yacht engine, ***author's***, 3½ and 8×6 fore and aft compound, for steamer ***Stornoway I***, ills., 1,072.
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